COMPUTER AIDED THERMAL HYDRAULIC DESIGN OF SHELL AND TUBE HEAT EXCHANGERS

A Thesis Submitted

in Partial Fulfillment of the Requirements

for the Degree of

MASTER OF TECHNOLOGY

By

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to the

NUCLEAR ENGINEERING AND TECHNOLOGY PROGRAMME

INDIAN INSTITUTE OF TECHNOLOGY KANPUR

MARCH 1990

NETP-1990-M-PAR-COM

5/3/90

CERTIFICATE

This is to certify that this work on 'Computer Aide Thermal Hydraulic Design of Shell and Tube Heat Exchangers' b Shiv Narayan Pareek has been carried out under my supervision an has not been submitted elsewhere for the award of a degree.

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DEDICATED TO MEMORIES OF

MY MOTHER.

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ACKNOWLEDGEMENTS

I thank Mr. S. C. Haldar, of Nuclear Power Corporation of India Ltd. for supplying relevant material and for the suggestion of this work.

I am very grateful to Dr. K. Sri Ram and Dr. P. Munshi for their guidance and encouragement during this work.

The help I received from my friends and colleagues during this work is gratefully acknowledged.

S. N. PAREEK

ABSTRACT

A menu driven interactive Pascal program has been developed for the thermal hydraulic design of shell and tube heat exchangers. The manufacturing tolerances and data for geometrical clearances are taken from TEMA for class R exchangers. The program can be used for rating as well as for the design purposes. The Kern method is used to get the primary design which is further modified by Bell method. Finally the checks of possible vibration damage because of vortex shedding and fluid elastic whirling in addition to the baffle and collision damage numbers are carried out.

The results are encouraging when checked for the avilable case studies, considering the complexity of the problem.

NOMENCLATURE

Subscripts s,t and m are used for shell side, tube side and tube metal properties respectively.

A	Surface area, m^2
A_{ba}	Bypass area of one baffle, m^2
A_{mb}	Minimum flow area at center line of one baffle, m^2
A_{sb}	Shell to baffle leakage area, m^2
A_{tb}	Tube to baffle leakage area, m^2
A_w	Net flow area through one window, m^2
A_{wg}	Gross window flow area, m^2
A_{wt}	Baffle window area occupied by tubes, m^2
B_c	Baffle cut, % of shell diameter
B_t	Baffle thickness, m
C_n	Frequency constant
C_p	Specific heat for constant pressure, J/(kg.K)
C_v	Specific heat for constant volume, J/(kg.K)
D_{ctl}	Pitch Diameter of outer most tube, mm
D_b	Baffle Diameter mm
D_i	Inside diameter of tube, mm
D_{otl}	Outer tube limit diameter, mm
D_s	Shell Diameter mm
D_t	Tube outside diameter, mm
D_w	Equivalent Hydraulic diameter of baffle window, mm
E	Modulus of elasticity of tube metal, N/m^2

F_c	Fraction of tubes in cross flow
F_w	Fraction of tubes in one window
f	Friction factor
f_n	Natural frequency, Hz
f_{vs}	Vortex shedding frequency, Hz
G	mass velocity, kg/(m^{2}sec.)
G_w	Window mass velocity kg/(m^{2}sec.)
g_c	Gravitational conversion constant, (in S. I. unit = 1)
h_i	Ideal Bundle heat transfer coefficient (W/m^{2}K)
h_s	Corrected shell side heat transfer coefficient (W/m^{2}K)
I	Moment of inertia, m^4
j	Heat transfer factor, given in appendix 1
J_c	Correction factor for baffle cut, to calculate h_s
J_L	Correction factor for baffle leakage effects
J_b	Correction factor for bundle by pass effects
J_r	Correction factor for adverse temperature gradients
J_s	Correction factor for variable baffle spacing
k	Thermal conductivity, (W/mK)
L	Length, m
L_b	Baffle spacing (L_{bi} = inlet, L_{bo} = outlet, L_{bc} = central), mm
L_{bb}	Bundle bypass diametral gap, mm
L_p	Bypass lane, mm
L_{pn}, L_{pp}	Tube pitch dimensions in appendix 1
L_{sb}	Diametral clearance, shell to baffle mm
L_{tb}	Tube to baffle diametral clearance, mm
L_{ts}	Thickness of Tubesheet, mm
N_b	Number of baffles

N_{ss}	Number of pairs of sealing strips
N_t	Total number of tubes in exchanger
N_{tcc}	Number of effective tube rows in cross flow
N_{tcw}	Number of effective tube rows in baffle window
R_b	Correction factor for bypass flows for for ΔP_s calculation
Re	Reynold Number (subscript $_s$ = shell , $_t$ = tube side)
R_l	Correction factor for leakage effects for ΔP_s calculation
R_s	Correction factor for inlet and outlet zone flow expansion
Sl_c	Strouhal number,
T	Temperature, $^{\circ}C$
U	Overall heat transfer coefficient, $W/(m^2K)$
V_c	Cross flow velocity, m/sec.
V_{crit}	Critical cross flow velocity, m/sec.

Greek symbols :

β	Instability constant used in Conors equation (3.4).
δ_o	Logarithmic decrement of amplitudes because of damping
μ	Dynamic viscosity of fluids, $N.s/m^2$
μ_w	Dynamic viscosity of fluid at wall temperature
ρ	Density kg/m^3
θ	Angles defined in fig.5, radian

INTRODUCTION

The use of computer programs is essential for thermal hydraulic design of heat exchangers as they enable to investigate rapidly all parameters to arrive at an optimum solution. Manufacturers use computer programs to design heat exchangers that will meet the required duty within certain operating constraints. Users, on the other hand use programs to check what a given exchanger will achieve if operated in a specified way.

This thesis is concerned with the development of a computer code for thermal hydraulic design of shell and tube heat exchangers. A brief introduction of shell and tube heat exchangers including their types, and criterion for selection and design are given in following sections 1.1 to 1.3.

1.1 Shell and Tube Heat Exchangers, Components and Designation :

Heat Exchangers built of round tubes mounted in cylindrical shell with their axes parallel to that of the shell are called Shell and Tube Heat Exchangers. It is the most common of the various types since it accounts for over 50% of all installed. Although it is not compact, it is robust and its shape makes it well suited to pressure operation. It is also versatile and it can be designed to suit any application. Except for the special purpose air cooled heat exchanger, it is usually the only type

1. Stationary Head—Channel
2. Stationary Head—Bonnet
3. Stationary Head Flange—Channel or Bonnet
4. Channel Cover
5. Stationary Head Nozzle
6. Stationary Tubesheet
7. Tubes
8. Shell
9. Shell Cover
10. Shell Flange—Stationary Head End
11. Shell Flange—Rear Head End
12. Shell Nozzle
13. Shell Cover Flange
14. Expansion Joint
15. Floating Tubesheet
16. Floating Head Cover
17. Floating Head Flange
18. Floating Head Backing Device
19. Split Shear Ring
20. Slip-on Backing Flange
21. Floating Head Cover—External
22. Floating Tubesheet Skirt
23. Packing Box
24. Packing
25. Packing Gland
26. Lantern Ring
27. Tierods and Spacers
28. Transverse Baffles or Support Plates
29. Impingement Plate
30. Longitudinal Baffle
31. Pass Partition
32. Vent Connection
33. Drain Connection
34. Instrument Connection
35. Support Saddle
36. Lifting Lug
37. Support Bracket
38. Weir
39. Liquid Level Connection

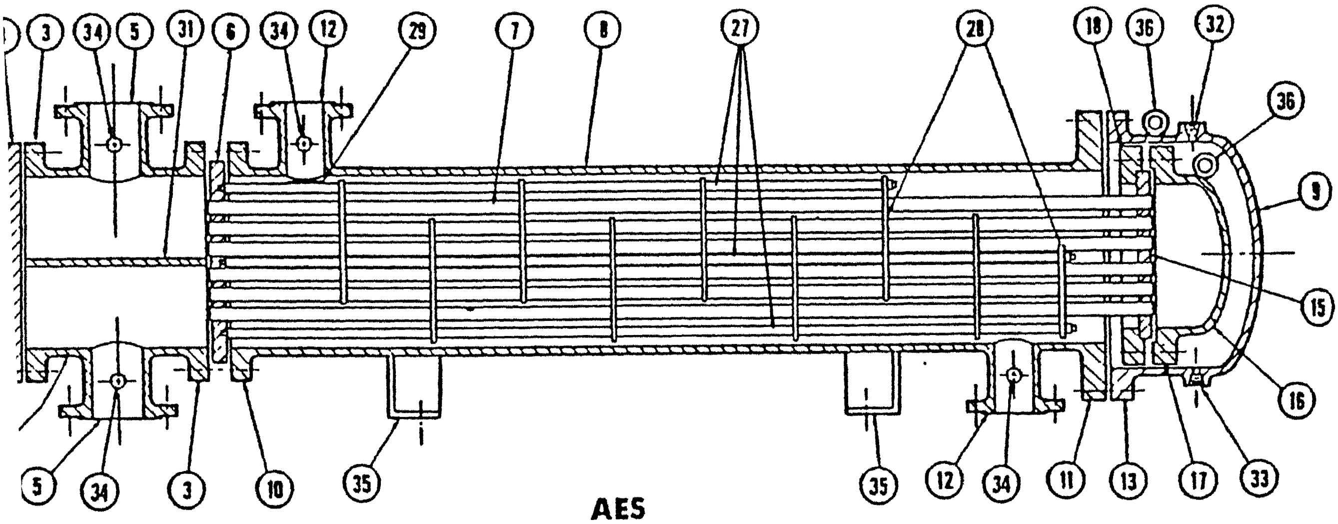

Fig.1. HEAT EXCHANGER, CONSTRUCTION AND TERMINOLOGY (TEMA).

which can be considered for large surface areas having pressures greater than 30 Bars and temperatures greater than 260 °C.

The nomenclature typically used in practice and Recommended by Tubular Exchanger Manufacturers Association, (**TEMA**) for heat exchanger components is shown in Fig.1.

Many variations of shell and tube exchanger type are available, depending upon the combinations of shell, and head (front and rear end head) types. The difference lies mainly in the detailed features of construction and provisions for differential thermal expansion between tubes and shell. The Shell and head types are described in section 1.1 to 1.2.

1.1.1 Head Types

TEMA classification for rear and front end heads is as shown in Fig.2. and described as under.

1.1.1.1 Channel type head (TEMA A,L,C,N)

This type of head is used when cleaning of inside of tubes is to be performed frequently, because it permits access to tube side without disturbing piping connections. It can be either integral with shell(TEMA C,N) or bolted to it (TEMA A,L).

1.1.1.2 Bonnet type head (TEMA B,M)

It is usually fitted at rear end of fixed tube sheet exchangers when nozzles are not there and cleaning of tube side is not frequent, because it permits access to tubes only after removing piping connections, but it is cheapest design.

1.1.2 Shell Types

Depending upon nozzle arrangement on shell side, there are seven TEMA standard arrangements as shown in the fig.2. The most

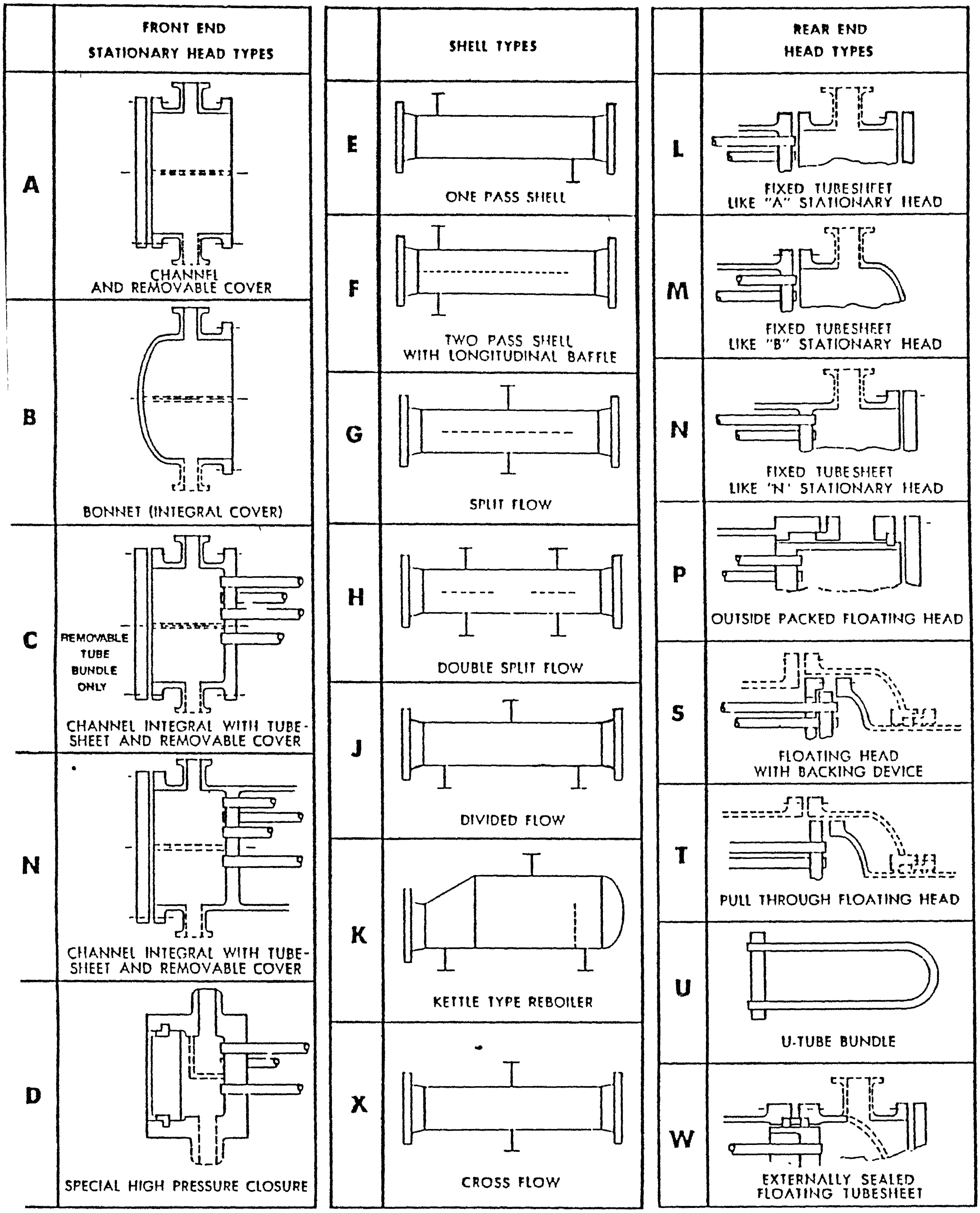

Fig.2. SHELL, FRONT END, AND REAR HEADS, TEMA CLASSIFICATION

common arrangement is E type shell which has one shell side pass. Other arrangements are F type which has two shell side passes and inlet and outlet nozzles on same end of shell is used, when it is needed to increase shell side flow velocity hence pressure drop with out increasing the length of shell. The split flow arrangements (type G, H, J) are used to reduce shell side pressure drop. Type K is used in kettle type reboilers used to vaporize the shell side fluid, and cross flow shell type X also has low pressure drop characteristics (lower than J type) because there are no segmental baffles so shell side fluid makes only one pass across the bundle.

In horizontal units inlet should be at the bottom and outlet at the top rather than both at the sides, This minimizes the bypass of shell side fluid.

1.1.3 Designation

TEMA Recommends that Heat exchanger type and size be designated by shell inside diameter in inches rounded to nearest integer followed by tube length in inches. For example,

SIZE 19-84 TYPE BGU is the designation for a U tube exchanger(TEMA U) with bonnet type stationary head(TEMA B), split flow shell(TEMA G), 19" inside diameter shell with tubes 7' straight length. Classification written in parentheses is that described by TEMA ,and is shown in Fig.2.

1.2 Principal types of shell and tube heat exchangers

1.2.1 Fixed Tube sheet (TEMA L, M, N)

As the name implies tube sheets are welded to shell, so thermal expansion is the criterion for its selection. Although it is an extremely leak proof design but shell side cleaning can only be done by chemical washing. Expansion joints in shell, if provided, reduces over-stressing of components because of differential thermal expansion. It is the least cost and easiest to fabricate type.

1.2.2 U tube (TEMA U) type

It needs only one tube sheet, and U tubes are free to expand relative to shell so problem of thermal expansion is eliminated and design is suitable for high pressure and temperature. Tube bundle is removable so mechanical means can be adopted to clean on shell side, but tube side cleaning becomes difficult because of bend. For clean tube side fluids with higher pressure and temperature operations this is ideally suitable.

1.2.3 Floating head type

In this type one tube sheet is fixed but other is free to float. There are 4 different types under this category, which are described below.

1.2.3.1 Split backing ring (TEMA S)

To separate the shell and tube side fluids at floating head end, the tube side of the floating tube sheet is fitted with a flanged, gasketed cover at its periphery which is held in position by bolting it to a split backing ring on the other side of the tube sheet. The complete floating head assembly is

located beyond the main shell in a shell cover of larger diameter. The nature of floating head construction limits it to an internal (tube side) design pressure of the order of 750 lb/in^2 (50 bar). Outer tube limit (OTL) is limited by the inside diameter of floating head gasket or inside diameter of backing ring whichever is smaller.

1.2.3.2 Pull through (TEMA T)

In this type floating head cover is directly bolted to floating tube sheet so its diameter is larger, which further increases the shell diameter approximately the same as the enlarged shell cover of the split backing ring type to contain same number of tubes. The important advantage is easier dismantling and tube bundle removal reducing downtime and maintenance. Shell bundle clearance is largest of all types.

1.2.3.3 Packed latern ring type or externally sealed type (TEMA W)

The separation of shell and tube side fluids at floating head is obtained by means of packing rings installed between the outside of the floating tube sheet and recesses in the rear head flanges. The shell and tube side fluids each have their own packing rings,which are separated by a latern ring provided with weep holes for leak detection. It is the least expensive removable bundle type exchanger. Generally used below 200 °C and 300 psi. Only one or two tube passes can be used.

1.2.3.4 Outside packed (TEMA P)

In this type shell side fluid is still contained by glanded packing as in externally sealed (TEMA W) type but tube side fluid

is contained by metallic seal of machined surfaces so there is no restriction of pressure, temperature and corrosiveness of tube side fluid.

1.3 Thermal Design Parameters and Selection Criterion

1.3.1 Tube Diameter

The selection of tube diameter depends upon fouling nature of fluid, space available and cost. Compact and economical units however are obtained by using small diameter tubes. Tubes of 3/4 and 1 in. are most widely used.

1.3.2 Tube Thickness

The tube thickness is designated by Bermingham Wire Gauge (BWG). The standard thicknesses for different nominal diameters of tubes are given by TEMA. The minimum tube thickness is selected considering the pressure differential across the wall, corrosiveness of fluids, flow induced vibration, Axial strength and standardization.

1.3.3 Tube Length

Tubes are selected as long as possible considering constraints of handling and space available at the time of bundle removal, because cheapest construction with given surface area will have smallest shell diameter. The maximum tube length for removable bundle exchangers is about 9 m with bundle weight approx. 20 tones. For fixed tube sheet exchangers the tube length can be up to 15 m. Tube lengths of 8,12,16,20 and 24 ft are used as standard lengths and recommended by TEMA.

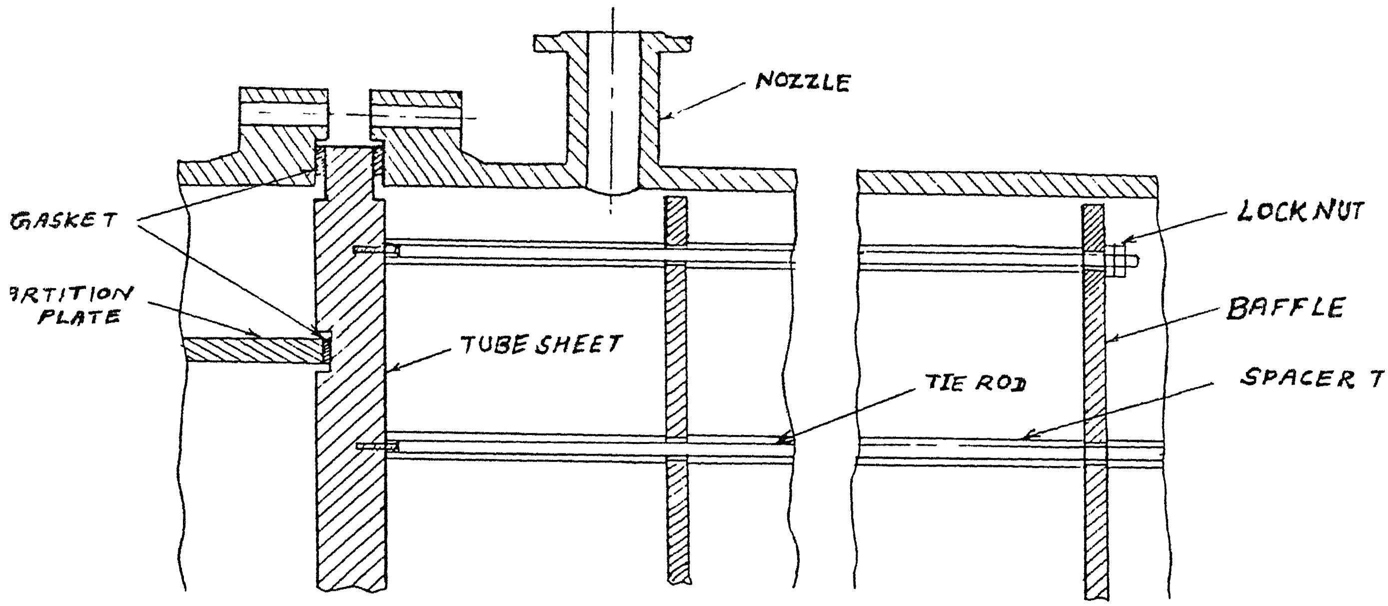

Fig.3. TIE RODS AND SPACERS

1.3.4 Tube Pitch and Pattern

Use of small pitch (tube to tube center distance) results in compact exchangers, but again this is limited by fouling nature of shell side fluid and tube to tube sheet welding technique used.

Square and diamond patterns of tubes permits external tube surface cleaning by mechanical tools, but triangular patterns will not so use is limited to clean service. Standard pitch to tube diameter ratios in use are 1.25, 1.33 and 1.5. Triangular pattern results in larger pressure drop and heat transfer coefficient on shell side.

1.3.5 Number of Tube Passes

Multiple tube passes are used to increase the tube side velocity so as to lower the rate of fouling and increase heat transfer coefficient within allowable pressure drop. Usually even number of tube passes are used because of mechanical difficulties associated with odd tube passes in providing tube fluid headers on both sides of exchangers(particularly floating head types). Generally up to 8 tube passes are used, because to provide passes the head is fitted with flat metal plates, (known as partition plates, shown in Fig.3.) which are welded to the tube sheet. This makes design less compact and difficult to access. Tube passes are arranged in such a way that the tube side can be drained and vented.

1.3.6 Special Tubes

Bi-metallic tubes are used when single metal is not compatible with both shell and tube side fluids. Low fin tubes (

in which the fins of nominal height 1.5 mm are extruded from the parent tube) are considered if shell side heat transfer coefficient is to be increased. Sometimes tube inserts are used if tube side heat transfer coefficient is to be increased.

1.3.7 Baffles

The use of baffles in shell and tube heat exchangers increases cross flow velocity thus heat transfer coefficient and reduces fouling. Baffles also act as tube supports against sagging and vibration. Different types are as follows-

(1) Orifice Baffles are circular disks having holes larger then tube outer diameter which permits the shell fluid to pass through. These can be used only with very clean shell fluids.

(2) Disc and Doughnut are essentially like orifice baffles but are cut into two portions a disk and other is called doughnut.

(3) Segmental Baffles are having baffle holes smaller than in orifice and disc - doughnut types.

The clearance between baffle hole and tube outer diameter is kept as small as possible so that tubes can be made to pass through. Segmental baffles can be single segmental, double segmental or triple segmental. Single segmental baffles give the maximum cross flow velocity,heat transfer coefficient and pressure drop. Double and triple segmental baffles are used when permissible pressure drop on shell side is smaller and designer can not afford to have single segmental baffles.

The cut portion of single segmental baffle is called window and specified by % of shell inner diameter. Smaller the baffle

cut more will be the amount of cross flow but pressure drop will also be more, also larger baffle cut will result in maldistribution of shell flow and small heat transfer coefficient. Baffle cut of 20 to 25 % is considered optimum.

Generally baffles are spaced at a distance smaller than or equivalent to shell diameter and more than one fifth of the shell diameter [ref.TEMA]. Baffle spacing influences the shell side flow velocity so lower limit for baffle spacing is limited by maximum allowable shell side pressure drop and performance loss resulting from increase in leakage streams. Upper limit for baffle spacing is due to reduction in heat transfer coefficient because of increase in parallel flow and increased susceptibility to flow induced vibration because of decreased stiffness of tubes hence less natural frequency. Distance between the adjacent baffles is generally kept equal except at the ends where it is generally more than central baffle spacing to accommodate inlet and outlet nozzles and impingement plate etc.

1.3.8 Routing of Fluids

Allocation of fluids to shell or tube side depends upon several factors *i.e.*

(1) Corrosive nature of fluid :-

more corrosive of the fluids is to be placed on tube side because it is easier to replace damaged tubes individually.

(2) Fouling nature :

Shell side is difficult to clean than tube side except in he U tube exchangers, so fouling fluid is routed to tube side.

(3) Fluid Temperature and Pressure :

It is economical and safer to contain the high pressure and temperature fluid in tubes than in shell.

(4) Allowable Pressure Drop :

For same amount of pressure drop higher heat transfer coefficient is obtained on tube side, so fluid with less allowable pressure drop is placed to tube side. Also prediction of pressure drop on the tube side is more accurate, so when it is crucial to limit pressure drop for a particular fluid, it is placed to tube side.

(5) Flow rate :

Because Reynold's number for turbulent flow on shell side is around 200, so smaller flow rate on shell side will experience more turbulence. On the other hand larger flow rate on shell side will require multiple tube pass construction with consequent temperature drop efficiency, so better overall design is obtained when smaller quantity fluid is placed on shell side.

(7) Hazardous or expensive fluid is to be placed on tightest side which is generally tube side.

(8) In the case of no restrictions as listed above, The arrangement which gives higher overall heat transfer coefficient with out exceeding allowable pressure drop is chosen.

1.4 OBJECTIVES OF STUDY

The main objective is to develop a simpler code than proprietary codes for the thermal-hydraulic design of shell and tube heat exchangers, which can also be used for the rating and flow induced vibration checking of existing heat exchangers

without the requirement of large memory and computing system. The software developed under this thesis is best suited to be used with Personal Computers, with at least 256 K RAM.

DESIGN METHODS AND LITERATURE SURVEY

2.1 General Classification

Three different types of approaches discussed in literature are as follows

2.1.1 Integral Approach

This approach considers total flow as effective in heat transfer and pressure drop calculations and introduces simple correction factors. The procedures based upon this approach are- Kern's method[12], and Donohue's method[7]

2.1.2 Semi Analytical Approach

This approach takes in to account the affect of non idealities and uses correction factors based upon the estimation of different streams. Bell Delaware method[1] is the example of this approach which is discussed in detail later.

2.1.3 Stream Analysis Method

This is the most complex approach proposed by Tinker.[24] This method is the basis of Heat Transfer Research Institute (HTRI) and Proprietary Computer programs. Several researchers have proposed their simplified methods based upon this approach.

Since later discussions refer to the various flow streams as designated by Tinker[24], a brief explanation is as follows-

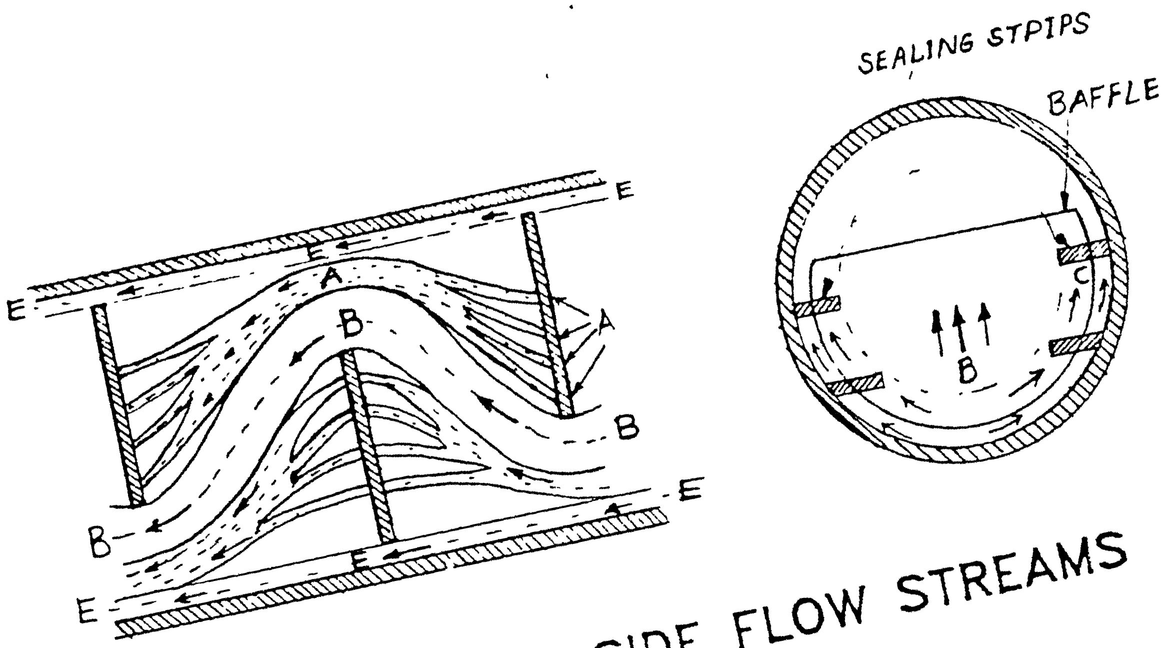

Fig.4A.SHELL SIDE FLOW STREAMS

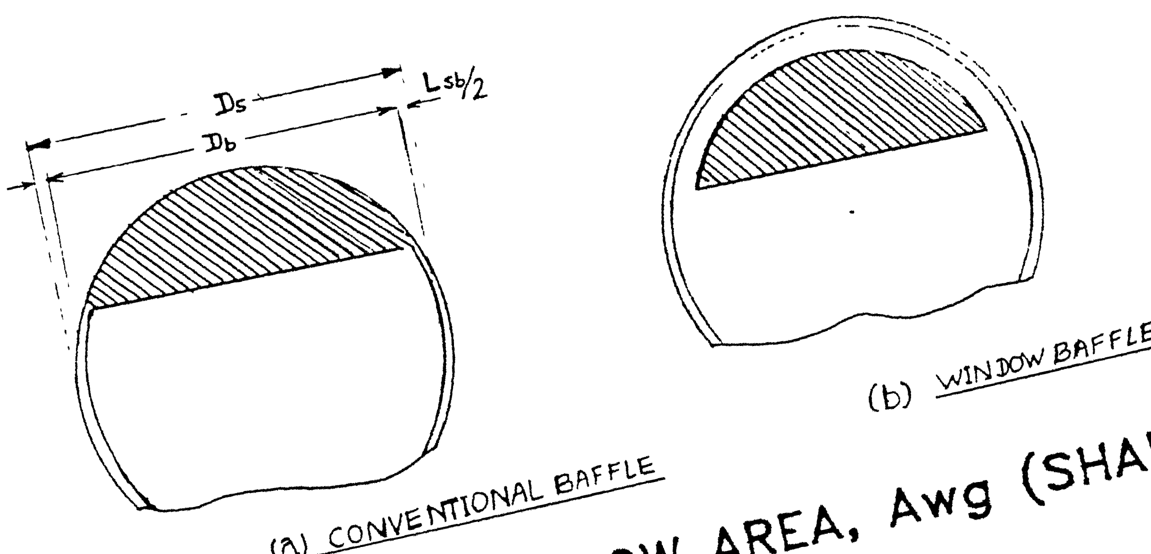

Fig.4B. WINDOW FLOW AREA, Awg (SHADED)

Tinker divided the shell flow into four streams. As shown in fig. 4A. A is the leakage stream through the tube-baffle clearance in one baffle, B is the cross flow stream through the bundle, C is the bypass stream between the shell and the outside of the tube bundle, and E is the leakage stream through the baffle-shell clearance. An F stream (not shown in fig.) is also defined for multiple tube pass exchangers, because of flow along the pass partition plate.

The relative effectiveness of the flow streams towards heat transfer is B > A > (C, F) > E. so C, F, E and A streams should be minimized in comparison of B stream. It is accomplished by -

(1) Using small manufacturing clearances.

(2) Using Sealing strips to minimize C stream particularly in floating head designs.

(3) By avoiding too small (< 0.2*Shell ID) baffle spacing.

(4) By avoiding laminar flow on shell side, C,F,and E streams can be minimized.

(5) Fraction of B stream reduces with reduction in shell diameter, so careful designing is needed.

3.2 Bell method [4]

The following section formulates the method, all equations are given in consistent units so no conversion factor is needed for S I units. The nomenclature is also given but unit is given at the beginning with complete nomenclature of all symbols used.

A simpler method was developed by Bell under the Delaware

University Research Program[1] and further refined by him and other modifications and improvements have been made by Taborek[22] mainly in the R_b and R_l curves and some changes in J_b and J_l data.

The method provide calculation procedure for shell side heat transfer and pressure drop.

The approach first calculates the ideal heat transfer and pressure drop on the basis of total flow across the bundle and then five correction factors are applied to approximate the effect of leakages and bypassing.

The basic equation is

$$h_s = h_i * J_c . J_l . J_b . J_s . J_r \qquad ....(2.1)$$

where h_i = Heat transfer coefficient for pure cross flow in an ideal tube bank assuming the entire shell side stream flows across the tube array.

J_c = correction factor for baffle cut. This correction includes the effect of window and bundle heat transfer. It has the value 1.0 for an exchanger with no tubes in window and increases to 1.15 for small baffle cuts and decreases to 0.65 for large baffle cuts. For a typical well designed heat exchanger the value is near 1.0.

J_l = correction factor for baffle leakage effects including both tube-baffle and baffle-shell leakage(A and E streams) with a heavy weight is given to latter. It is a function of the ratio of leakage to cross flow area and a function of the clearances. A typical value of J_l is in the range of 0.7 to 0.8.

J_b = correction factor for bundle bypass (C and F streams). For a fixed tube sheet construction J_b =~ 0.9 and for a pull through J_b =~ 0.7 which can be increased to 0.9 by the use of sealing strips.

J_s = correction factor for variable baffle spacing at the inlet and outlet sections. The nozzle locations result in larger spacing and lower velocities and thus lower heat transfer coefficients. J_s usually ranges from 0.85 to 1.0.

J_r = correction factor for any adverse temperature gradient buildup in laminar flows. This correction applies only for shell side Reynolds Numbers of less than 100.

Shell side pressure drop is the summation of pressure drops in the inlet and outlet sections and the baffle central section, consisting of the cross flow and window pressure drop. The following correction factors are used.

R_l = Correction for leakage (A and E streams). This correction is based on the same factor as J_l but is of different magnitude. Usually R_l =~ 0.4 to 0.5 although lower values are possible with small baffle spacing.

R_b = Correction factor for bypass flow (C and F streams). It is different in magnitude from J_b and ranges from 0.5 to 0.8.

R_s = Correction factor for inlet and outlet sections having a baffle spacing different from central section.

Before the correction factors can be determined it is necessary to calculate various leakage and flow areas. These

auxiliary calculations are based upon the geometry of baffle, and given in appendix (A).

2.2.1 Shell side heat transfer coefficient

To calculate the heat transfer coefficient the step wise procedure is as given below

(1) Maximum mass flow velocity, G_s

$$G_s = \frac{W_t}{A_{mb}} \qquad ....(2.2)$$

where W_t is total flow rate, and A_{mb} is calculated and defined in Appendix A, section (j),equation(A.17).

(2) Shell side cross flow Reynolds number, $Re_s = \dfrac{D_t . G_s}{\mu_s}$ (2.3)

where D_t is tube diameter and μ_s is Dynamic viscosity of shell fluid.

(3) Ideal heat transfer coefficient for cross flow over bundle is given by

$$h_i = j . c_p . G_s . Pr_s^{-2/3} . \phi_s \qquad ....(2.4)$$

where Pr_s is Prandtl number based on property values at average fluid temperature, c_p is specific heat of shell fluid, ϕ_s is viscosity correction factor,

$$\phi_s = (\mu_s/\mu_w)^{0.14} \qquad ....(2.5)$$

μ_w is dynamic viscosity of shell fluid at tube wall temperature. For a nonviscous fluid such as water on shell side ϕ_s can be taken unity,

j is heat transfer factor defined as , $j = St . Pr^{1-n}$

where St is Stanton number and n is index of Prandtl number in the correlation of the type

$$Nu = C . Re^m . Pr^n$$

Value of heat transfer factor j is reported in literature, For the present work it has been taken from[48] charts prepared by Engineering Sciences Data Unit International Ltd. (ESDU), and fitted in the form of logarithmic polynomials,

$\ln j = f(\ln Re)$ given in appendix B.

(4) Having calculated F_c (Eq A.7), appendix A, Segmental baffle window correction factor J_c can be calculated from polynomial of the form, $J_c = f(F_c)$, given in appendix B.

(5) Using values calculated in sections (h), (i), (j) of Appendix A, The following ratios can be calculated.

$$r_{lm} = \frac{A_{sb} + A_{tb}}{A_{mb}} \qquad \ldots\ldots.(2.6)$$

and

$$r_s = \frac{A_{sb}}{A_{sb} + A_{tb}} \qquad \ldots\ldots.(2.7)$$

The baffle leakage correction for heat transfer, J_l is determined from polynomial of form, $J_l = f(r_{lm})_{r_s}$, given in Appendix B.

Similarly baffle leakage correction for pressure drop, R_l is determined from polynomial $R_l = f(r_{lm})_{r_s}$, given in Appendix B

(6) Using Equations (A.14), (A.17), and (A.11) from appendix A, Following ratios are computed.

$$r_b = \frac{A_{ba}}{A_{mb}} \qquad \ldots\ldots.(2.8)$$

$$N_{ss}^{+} = \frac{N_{ss}}{N_{tcc}} \qquad \ldots\ldots.(2.9)$$

where N_{ss} is number of pairs of sealing strips, shown in (fig 4A). Bundle bypass correction factor for heat transfer J_b is then calculated from the polynomial of the form,

$J_b = f(r_b)_{N_{ss}^{+}}$ given in Appendix B.

Similarly bundle bypass correction factor for pressure drop R_b is

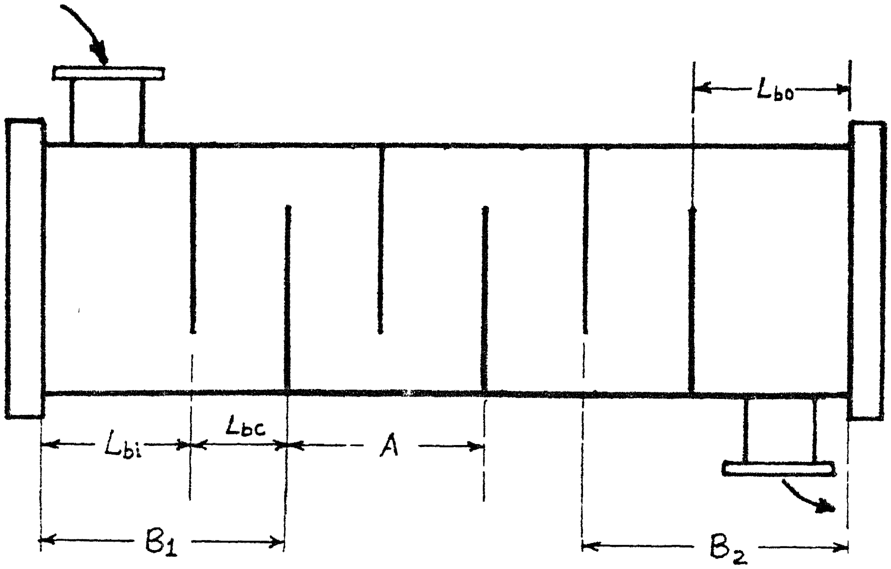

Fig.5A. BAFFLE ARRANGEMENT

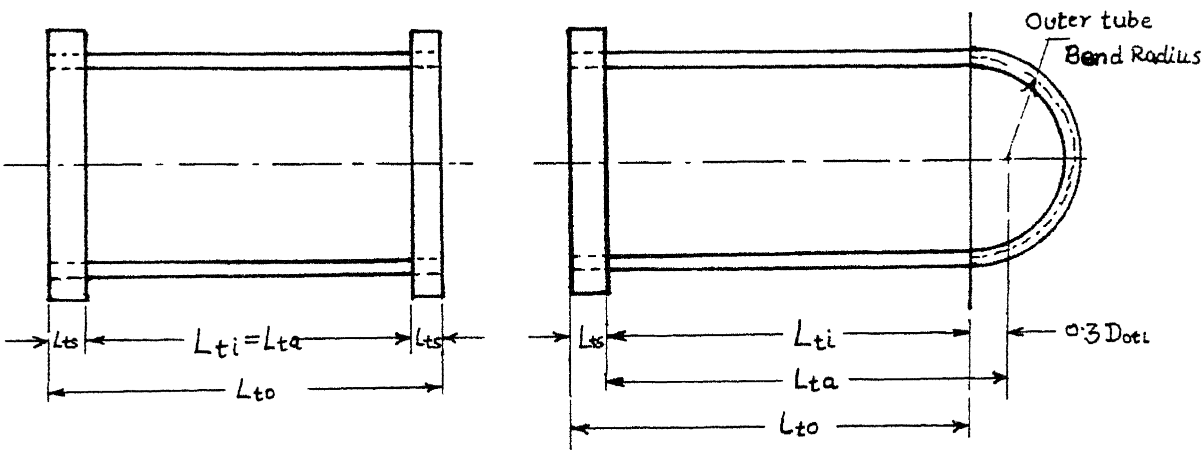

Fig.5B. TUBE LENGTH DEFINITIONS

calculated from polynomial of type, $R_b = f(r_b)_{N_{ss}^+}$, given in Appendix B.

(7) If shell side Reynolds number is less than 100, (flow is laminar) then adverse temperature correction J_r is to be estimated. This is a function of the number of tube rows crossed as calculated,

$$N_c = N_{tcc} + N_{tcw} \qquad \text{.......(2.10)}$$

where N_{tcc} and N_{tcw} are calculated in Appendix A, and

$$J_r = \begin{cases} 1 & \text{for } Re_s > 100 \\ \left(\frac{10}{N_c}\right)^{0.18} & \text{for } Re_s <= 20 \end{cases} \qquad \text{.......(2.11)}$$

and a linear interpolation for $20 < Re_s < 100$.

(8) Heat transfer correction for unequal end spacing , J_s is determined from,

$$J_s = \frac{(N_b - 1) + (L_i^+)^{1-n} + (L_o^+)^{1-n}}{(N_b - 1) + L_i^+ + L_o^+} \qquad \text{.....(2.12)}$$

where n = 0.6 for turbulent flow.

For laminar flow correction factor J_s is about $(1+J_s)/2$.

where $L_i^+ = L_{bi}/L_{bc}$ and $L_o^+ = L_{bo}/L_{bc}$,

L_{bi} is inlet baffle spacing, L_{bo} is outlet spacing and L_{bc} is central baffle spacing as shown in fig.5A.

The fig.5A. shows maximum unsupported tube span in central region is A, and in the inlet/outlet region it is B_1 and B_2.

(9) Having determined all correction factors, shell side heat transfer coefficient is calculated by using equation 2.1.

$$h_s = h_i . J_c . J_l . J_b . J_r . J_s \qquad \text{.......(2.13)}$$

2.2.2 Shell side pressure drop

The shell side pressure drop from this method is sum of cross flow pressure drop, ΔP_c, the baffle window pressure drop ΔP_w , and the pressure drop in inlet and outlet zones ΔP_e.

(1) Bundle pressure drop for ideal cross flow

$$\Delta P_{bi} = \frac{2.f.N_{tcc}.G_s^2.\phi_s}{g_c.\rho_s} \qquad ....(2.14)$$

where f is the cross flow friction factor, g_c is gravitational constant (for SI unit g_c = 1).

(2) Cross flow pressure drop,

$$\Delta P_c = \Delta P_{bi}(N_b - 1)R_b.R_l \qquad ....(2.15)$$

where N_b is number of baffles, eq (A.13),Appendix A, and factors R_b and R_l are determined from polynomials listed in Appendix B

(3) Window pressure drop , ΔP_w For turbulent flow (Re_s > 100)

$$\Delta P_w = \frac{N_b(2 + 0.6*N_{tcw})G_w^2.R_l}{2.g_c.\rho_s} \qquad ....(2.16)$$

where G_w is window mass velocity given as

$$G_w = \frac{W_T}{\sqrt{(A_{mb}.A_w)}} \qquad (2.17)$$

For laminar flow (Re_s < 100),

$$\Delta P_w = N_b\left\{26.\frac{G_w.\mu_s}{\rho_s}\left[\frac{N_{tcw}}{L_{tp} - D_t} + \frac{L_{bc}}{D_w^2}\right] + 2(10^{-3})\frac{G_w^2}{2\rho_s}\right\}R_l \qquad (2.18)$$

(4) End zone pressure drop,

$$\Delta P_e = \Delta P_{bi}\left(1 + \frac{N_{tcw}}{N_{tcc}}\right)R_b.R_s \qquad ....(2.19)$$

where

$$R_s = \left(\frac{L_{bc}}{L_{bo}}\right)^{2-n} + \left(\frac{L_{bc}}{L_{bi}}\right)^{2-n} \qquad ....(2.20)$$

where n = 0.2 for turbulent flow and 1.0 for laminar flow.

(5) Shell side pressure drop

$$\Delta P_s = \Delta P_c + \Delta P_w + \Delta P_e \qquad \text{.......(2.21)}$$

Pressure drop in inlet and outlet nozzles is not included here.

2.3 Kern method[12]

To begin with Bell's method, it is needed to have heat exchanger specifications so that can be used for rating purpose. We can use Kern method to design heat exchanger, then this design is further checked and modified by Bell method, which is considered more accurate.

2.3.1 Shell side pressure drop

Shell side pressure drop is proportional to number of times fluid crosses the bundle, for N number of baffles,

number of crosses = N+1 = Tube length /Baffle spacing,

considering equal baffle spacing.

The pressure drop on shell side can be written as

$$\Delta P_s = \frac{f_s . G_s^2 . D_s(N+1)}{2. g_c . \rho_s . D_e . \phi_s} \qquad \text{.......(2.22)}$$

where f_s is shell side friction factor

G_s shell side mass flow velocity

$$G_s = \frac{W_s}{A_s} \quad \text{and,}$$

$$A_s = \frac{D_s . (Pitch-D_o) . L_{bc}}{Pitch}$$

D_s shell inner diameter

g_c gravitational constant, for S I unit g_c = 1

ρ_s shell side fluid density

ϕ_s is correction for change in viscosity of fluid in boundary layer, For nonviscous fluids and turbulent flow ϕ_s = 1, and D_e is hydraulic equivalent diameter for shell side flow.

For this method the equivalent diameter D_e is calculated considering the shell side flow parallel to the tubes. Though this method does not distinguish between relative percentage of right angle flow to axial flow.

For 90,45 deg layout angles $$D_e = \frac{4.(P_T^2 - \pi D_t^2/4)}{\pi D_t}$$

For 30,60 deg layout angles $$D_e = \frac{4.(P_T^2 * \cos 30/2 - \pi D_t^2/8)}{\pi D_t}$$

2.3.2 Tube side pressure drop

Tube side pressure drop is calculated by the following equation

$$\Delta P_t = \frac{f_t . G_t^2 . L . N_{tpass}}{2 . g_c . \rho_t . D_t . \phi_t} \qquad ....(2.23)$$

where f_t is tube side friction factor

G_t is tube side mass flow velocity

$$G_t = \frac{N_t . \pi . D_t^2}{4 . N_{tpass}} \qquad ....(2.24)$$

N_t is Total number of tubes in exchanger

N_{tpass} is number of tube passes in the exchanger

L is length of tubes to be used in exchanger

ϕ_t is tube side viscosity correction = 1 for nonviscous fluids.

Shell and tube side friction factors are given by Kern as the function of shell and tube side Reynolds numbers Re_s and Re_t respectively, which were fitted into polynomial for use in

computer program.

$$Re_s = \frac{D_e . G_s}{\mu_s} \qquad ....(2.25)$$

$$Re_t = \frac{D_t . G_t}{\mu_t} \qquad ....(2.26)$$

where μ_s and μ_t are viscosity of shell and tube side fluids respectively.

2.3.3 Shell side heat transfer coefficient

The shell side heat transfer coefficient is calculated by the correlation suggested by Kern

$$h_s = J_h . \frac{k_s}{D_e} \left(Pr_s\right)^{1/3} . \phi_s \qquad ....(2.27)$$

where J_h is heat transfer factor given as a function of shell side Reynolds number, which has been fitted into polynomials for use in program, The Reynolds number used for estimation of J_h is same as used to calculate pressure drop.

Pr_s is Prandtl number of shell side fluid, $Pr_s = \frac{\mu_s . Cp_s}{k_s}$

2.3.4 Tube side heat transfer coefficient

For tube side also correlation given by Kern for water on tube side is used

$$h_t = J_{ht} . \frac{k_t}{D_t} \left(Pr_t\right)^{1/3} . \phi_t \qquad ....(2.28)$$

where J_{ht} is heat transfer factor for tube side flow given as a function of Reynolds number, k_t is thermal conductivity of tube side fluid, Pr_t is Prandtl number of tube side fluid and ϕ_t is viscosity correction factor for change of fluid viscosity, ϕ_t is taken = 1 for water.

2.4 Overall Heat Transfer Coefficient

Design Overall Heat Transfer Coefficient U_d is given as

$$\frac{1}{U_d} = \frac{1}{h_s} + R_{fo} + \frac{D_t}{2.K_m} \cdot \ln(\frac{D_t}{D_i}) + R_{fi}\frac{D_t}{D_i} + \frac{1}{h_t}.\frac{D_t}{D_i} \qquad ......(2.29)$$

where h_s is heat transfer coefficient for outside of tubes,(i.e. on shell side), R_{fo} and R_{fi} are fouling resistances on shell side and tube sides respectively, Δx is thickness of tube wall, K_m is thermal conductivity of tube metal, D_t and D_i are Outer and inner diameters of tube respectively.

Clean Overall Heat Transfer Coefficient, U_c is given by following equation

$$\frac{1}{U_c} = \frac{1}{h_s} + \frac{1}{h_t}.\frac{D_t}{D_i} \qquad ....(2.30)$$

neglecting the fouling factors on both sides,

Total Fouling Resistance also called Dirt Factor R_d,

$$R_d = R_{fo} + R_{fi}.\frac{D_t}{D_i} \qquad ....(2.31)$$

so using equations (2.29 to 2.31), $R_d = \dfrac{U_c - U_d}{U_c.U_d}$ ....(2.32)

2.5 Design Considerations

The Design Equation is $Q = U_d.A.\Delta t_m$ ....(2.33)

where Q is amount of heat to be transferred, which is determined from the requirement of temperature change of fluids.

U_d is Design overall heat transfer coefficient.

A is total Heat transfer surface area, which is total outer surface area of all tubes, which is to be determined.

Δt_m is mean temperature difference between shell and tube side fluids. For shell and tube heat exchangers with multiple tube

passes is determined by first estimating logarithmic temperature difference for a single pass counter flow exchanger and then using a correction factor F_T for the configuration under consideration. The value of F_T is always less than 1, given in the form of charts as a function of fluids temperature differences at ends and of the flow arrangements.

The reason why logarithmic mean temperature difference is used and not the arithmetic mean is because arithmetic temperature difference is independent of the flow pattern (*i.e.* both fluids flowing parallel or counter wise or across) and remains same for all patterns.

As shown by the design equation the compact design can be achieved by maximizing mean temperature difference and overall heat transfer coefficient. The counter flow arrangement gives the maximum mean temperature difference but this can not be used in all cases because of the following reasons-

(1) The tube lengths that can be used are limited by space available and handling difficulties.

(2) similarly the maximum shell size is also limited because of cost of manufacturing, transportation and reliability.

(3) Minimum tube side flow velocity is also limited to reduce fouling on tube side.

so cost of manufacturing and operating counter flow exchangers is the prohibiting factor. so are the shell and tube units with partially cross, parallel and counter flows in use.

The usual procedure to get mean temperature difference is by the use of analytical procedures. Expressions for F_T are

available for 2 and 4 passes of tube in single shell[25], but are limited in accuracy because of assumptions used for deriving, like -

Temperature at a cross section of shell and tube exchanger is constant, fluid property variation ignored,and number of tubes in all passes is assumed same, are a few to list.

A procedure has been written considering variation of number of tubes in different passes of a two pass shell and tube exchanger, in which fluid property variation is also possible. The method uses a piece wise integration of fundamental equation along the length of exchanger considering overall heat transfer coefficient constant along the length, thus yielding the overall heat transfer coefficient needed to achieve the given heat load (The amount of flow, and end temperatures of fluids) with out actually calculating the mean temperature difference. The program has also the choice to calculate amount of area needed to achieve the specified heat transfer and overall heat transfer coefficient.

Since the use of logarithmic mean temperature and correction for flow geometry,F_T are used in design extensively and TEMA also provides the charts for F_T, it is also used as a check in program based upon Kern method. It is recommended[12] that value of F_T should not be less than 0.75 and also used as an criterion to use more number of shell passes than one. The reason is that values of F_T below 0.75 are in a steep region so with a slight operational transient it may change considerably to affect exchanger performance. Value of F_T increases with increase in

number of shell passes, but cost and pressure drop also increases.

When total area for heat exchanger, A is obtained in equation (2.33), total number of tubes hence the diameter of shell is determined for given tube diameter , pitch, length and layout angle. The program uses data given by Kern and Krause[18] for estimating shell diameter for given tube passes.

The program TSHEET can also be used to calculate the full tube count for given shell diameter, tube diameter, tube pitch and layout angle.

TUBE COUNT

The number of tubes which can be accommodated within a given shell inside diameter is called tube count. It depends upon several design factors.

(1) Type of Exchanger :

Floating head type exchangers have Outer Tube Limit (OTL) smaller than fixed tube sheet type exchangers, so tube count will be smaller for same parameters (*i.e.* pitch, tube diameter etc.).

(2) Shell Side Flow :

Shell side flow determines the nozzle diameter and whether impingement plate is needed or not. Greater the nozzle diameter smaller the tube count, so as to increase tube count externally fitted impingement plate or distributor should be used.

(3) Number of Tube Passes:

As Tube passes are increased area occupied by pass partitions increases so tube count decreases.

(4) Tie rods, Spacers and Sealing devices also reduces the tube count.

(5) Tube End Attachment :

Welded ends need larger margins, which decreases tube count.

(6) In some cases tube sheets may have radiussed hubs at the periphery for welding to the shell or head barrel, so as tubes not to encroach radii, OTL is reduced.

(6) Rotatable bundles with internally fitted impingement plates requires two escape areas on opposite sides of the bundle.

So it is not possible to provide exact tube count. The usual procedure is to estimate full tube count for given diameter and then considering above mentioned factors and design practice the full tube count is reduced appropriately.

The Interactive Graphics Program TSHEET is prepared to calculate full tube count for given -

tube sheet diameter

margin from outer periphery

tube outer diameter

tube spacing or pitch

layout angle, and

location (coordinates) of any one starting tube.

The last variable makes the program very flexible that it is not necessary to have tube at the center, It can be forced to have tube at the location of designers choice with other variables remaining same. This feature is particularly important while designing for minimum leakage streams.

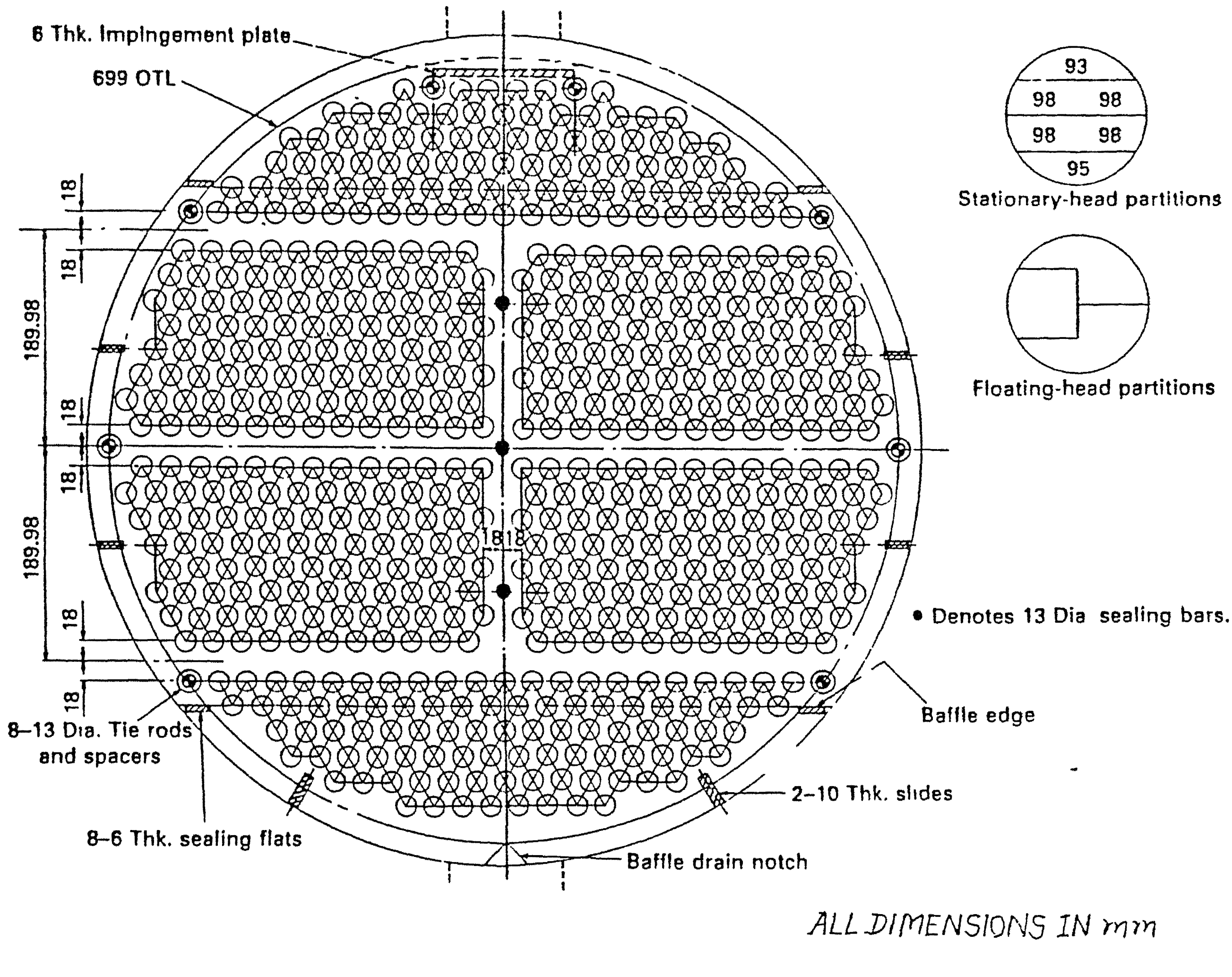

Fig. 6. THE ACTUAL ARRANGEMENT OF TUBES FOR 6 TUBE PASS, TEMA–S EXCHANGER, SHELL INSIDE DIAMETER 740 mm, TUBE OD 19.05 mm, PITCH RATIO 1.33, LAYOUT ANGLE 30

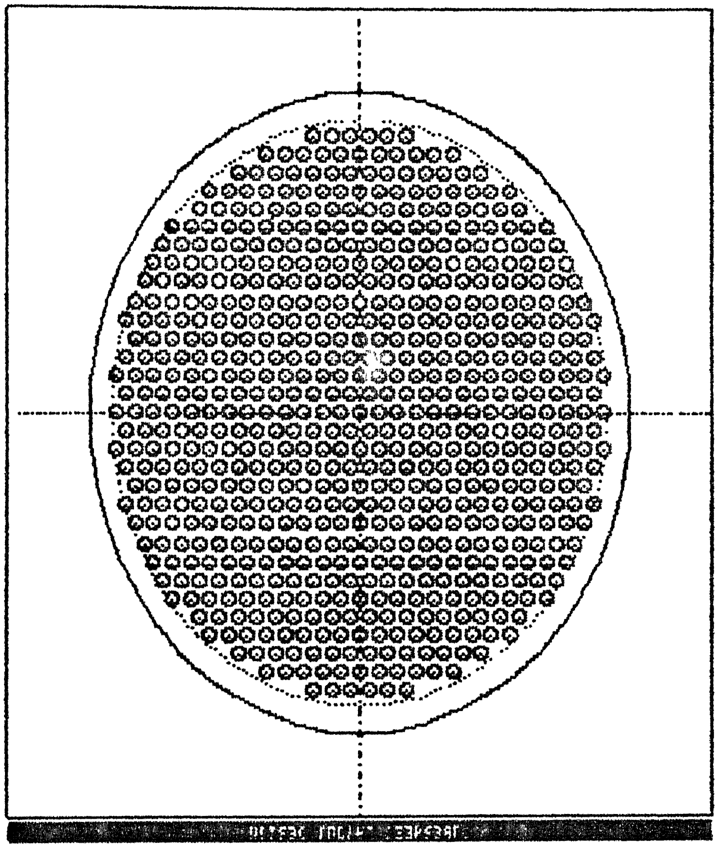

Fig.7. TOTAL TUBE COUNT = 649

since there is no direct procedure of getting actual tube count, the drawing of tube sheet layout with full tube count assists a lot in estimating of where tubes can not be installed, thus arriving at the actual tube count.

The Fig.7. shows a computer produced drawing of a tube sheet layout with full tube count and fig.6. shows the actual arrangement for a 740 dia shell, 6 pass, 580 tubes, 19.05 tube dia, for 25.4 pitch with 30 deg layout angle.

FLOW INDUCED VIBRATION

3.1 Reasons of Inclusion

As a result of improved pressure drop predictability, allowing the use of higher fluid velocities the number of exchangers experiencing vibration are increasing. Higher fluid velocities and improper design are major causative factors.

Flow induced vibration causes tubes to fail by following mechanisms-

[1] Fatigue due to repeated bending.

[2] 'Collision damage' because of repeated impact between adjacent tubes; which leads to the thinning of tube wall.

[3] 'Baffle damage' because of repeated impact between tubes and baffles,baffle tube clearance increases and tube wears out rapidly if baffle metal is harder than that of tube.

[4] Cutting at the tube-hole edges at the inner tube sheet face due to repeated impact between tube and tube sheet. This also results in failure of tube-tube sheet joint.

The majority of failures have been found in regions of long unsupported tube spans and high local velocity,such as inlet

baffle space (tubes in first baffle window). Analysis of failures suggested that had unsupported tube lengths been restricted to about 80% of TEMA values, few failures would have occurred. The effect of reducing the unsupported tube length to 80% of its initial value increases the tube natural frequency by 56%.

3.2 Tube Natural Frequency

To predict the resonant conditions leading to damage it is desirable to estimate the tube natural frequency, which depends upon the type of supports and span length, is given as

$$f_n = 0.04944\ C_n\ [E.I.g_c/(W_{eff}.L^4)]^{0.5} \qquad \text{.....(3.1)}$$

where f_n is natural frequency, E is modulus of elasticity of tube material, I sectional moment of inertia, L length of each span, C_n frequency constant (dimensionless), W_{eff} effective mass per unit length of tube, g_c gravitational constant ($g_c = 1$ in SI unit). Weff is the sum of mass per unit length of tube itself, mass per unit length of tube side fluid and virtual mass per unit length for the shell side fluid displaced by the tube. The virtual mass displaced is always more than actually displaced mass by a multiplier called 'Added mass coefficient' ($M_a > 1$). The effect of axial stress upon natural frequency is not considered by above equation.

For U bends the natural frequency is estimated by using longest bend length in above equation and then

$$f_{nu} = 0.829\ f_n \qquad \text{.....(3.2)}$$

which is the frequency for out of plane vibrations.

Values C_n and M_a are taken from[48]. For more than 10 spans

constant C_n is almost independent of mode and $\cong 32$. M_a is about 2 to 3 depending upon pitch and layout of tubes.

3.3 Excitation Mechanisms

Several different theories are used to evaluate the potential for the vibration problem. These can be classified as follows-

Classification of Flow Induced Vibration Mechanisms

- Cross Flow
  - Fluid Dynamic Excitation
    - Vortex Shedding
    - Turbulent Buffeting
    - Acoustic Excitations (When gas in shell)
  - Motion Dependent Excitation
    - Fluid Elastic Whirling
- Parallel Flow
  - TurbulentBuffeting

Vibrations induced because of parallel flow have amplitude much smaller so not considered damaging. The mechanisms and methods important for liquid (single phase) service are described under in some detail.

3.3.1 Vortex Shedding

When a bluff body is exposed to a perfectly uniform, nonturbulent incident flow above some critical velocity, the wake

behind the body develops instability, which oscillates in a regular sinusoidal pattern with an increase in velocity, and discrete vortices shed alternatively from the sides of the body, called 'vortex shedding'. This produces alternating lift and drag forces on the tube. The frequency of the alternating force in the drag direction is twice that in the lift direction, but of smaller amplitude. The vortex shedding frequency in lift direction (f_{vs}) is as given, by equation 3.3. When f_{vs} approaches natural frequency it results in large amplitude vibration, called 'Resonance'.

$$f_{vs} = Sl_c . V_c / D_o \qquad .....(3.3)$$

where Sl_c = Strouhal number, V_c = cross flow velocity based on minimum transverse area between tubes in a row, D_o = tube outside diameter.

When tubes vibrate at, or near, vortex shedding frequency, vortices shed at the tube natural frequency which is not given by the above equation. This feature is called 'lock-in' effect, which can prevail at a frequency of about 0.7-1.3 of the tube natural frequency. Therefore design criterion is to keep

$$f_n / f_{vs} < 0.5.$$

3.3.2 Turbulent Buffeting

Unsteady forces develop on a body exposed to highly turbulent flows, having a wide band of frequencies around a dominant central frequency which increases with the flow rate.

No data or equations are available for turbulent buffeting frequencies for liquids.

3.3.3 Fluid elastic whirling

It is a tube motion induced excitation mechanism. A momentary displacement of one tube in an array from its normal position alters the flow field, this disturbs the force balance on neighboring tubes and exposes them to a fluctuating pressure and causes them to change their positions in a vibratory manner at their natural frequencies. Thus this flow induced vibration originates from fluid elastic coupling. Tube motion may begin by vortex shedding or turbulent buffeting and become a catalyst for fluid elastic vibration.

When fluid velocity is less than a critical velocity excitation forces are balanced by fluid damping. Critical cross flow velocity can be calculated by the empirical equation 3.4.

$$V_{crit} = \beta . f_n . D_o . \sqrt{\frac{W_{eff} . \delta_o}{\rho_s . D_o^2}} \qquad \text{.....(3.4)}$$

where V_{crit} is critical cross flow velocity, β instability constant, δ_o logarithmic decrement, D_o tube OD and ρ_s shell fluid density. δ_o = 0.1 for liquids, and 0.03 for gases, W_{eff} is defined in section 3.2.

3.4 Damage Numbers

Thorngren[23] proposed two dimensionless numbers to be used to predict damage due to cutting action at the baffle or to collision between tubes.

$$N_{BD} = \frac{D_o . \rho_s . V_c^2 L^2}{F_B . S_f . g_c . A_m . B_t} \qquad \text{.....(3.5)}$$

$$N_{CD} = \frac{0.625\ D_o.\rho_s.V_c^2 L^4}{F_B^4 A_m(D_o^2+D_i^2)C_t.E.g_c} \qquad \ldots\ldots(3.6)$$

N_{BD} is baffle damage number and N_{CD} is collision damage number. where F_B = tube to baffle clearance factor, dimensionless. F_B = 1.00 for 1/32 in clearance, 1.25 for 1/64 in clearance. S_f = fatigue stress, C_t = minimum gap between adjacent tubes, A_m = cross sectional area of tube metal, D_o and D_i are outer and inner diameter of tube, V_c = cross flow velocity, ρ_s is shell fluid density, L = tube span length and g_c is gravitational constant (= 1 in SI unit).

The damage numbers are indicative of damage which vibration might cause, and Thorgren[23] stated that this would be avoided if both damage numbers are less than unity.

Erskine and Waddington[8] modified the above design criterion by studying case histories, that damage would be avoided if:

$$N_{BD} \text{ and } N_{CD} < \frac{0.00429\ \rho_s^{0.47}}{\mu_s^{0.235}} \qquad \ldots\ldots(3.7)$$

For vibration calculations cross flow areas are based on the free area between tubes in same row. The cross flow areas are therefore identical for those for heat transfer and pressure loss calculations for 30 and 90 pitch angles, but differs for 45 and 60 pitch angles.

3.5 Design procedure

The step wise procedure used by the computer program VIBUNIT to predict the possibility of damage because of flow induced vibration is as described under.

1. Calculate the tube natural frequency by using equation 3.1 for longest unsupported span of tube. . The longest unsupported spans for inlet, outlet and central regions are B_i, B_e, A as shown in figure 5A.

2. Calculate vortex shedding frequency by equation 3.3.

3. If ratio of f_{vs} to f_n is less than 0.5, the design is safe from the vortex shedding vibrations, otherwise corrective measures include the guide lines given in next section to make the tubes stiff or to reduce shell flow velocity.

4. Calculate critical cross flow velocity given by equation 3.4. If maximum cross flow velocity is smaller than critical cross flow velocity the design is safe.

5. Calculate Baffle, Collision and maximum damage numbers from equations 3.5 to 3.7. For safe design Baffle and Collision damage numbers should be smaller than maximum damage number or 1 whichever is smaller.

3.6 Design Recommendations for vibration prevention

Flow induced vibrations can be prevented by

[i] Increasing tube natural frequency, and

[ii] Reducing cross flow velocity.

Following are the recommendations-

[1] A cross flow shell(X type) or a no tube in window design virtually guarantees freedom from vibration failure.

[2] If damage is predicted near first row of tubes at inlet the problem can be solved by installing a support plate in the inlet baffle space.

[3] Shell side flow velocity can be reduced by using double or triple segmental baffles or by increasing shell side passes or by the use of wider pitch and pitch angle.

[4] U bend can be supported or all tubes at U bend can be clamped together to give more stiff design.

[5] Reduce baffle tube clearance and use baffle material softer than that of tubes,and increase baffle thickness.

[6] New design concepts such as Rod baffles or NEST supports can be considered.

PROGRAM DESCRIPTION

The Pascal programming language has been used in all implementations of this project. There are two executable files, namely HED.EXE and TSHEET.EXE. This chapter describes the uses, available features and limitations of both files.

The following four routines are available from the main menu driven program, HED.EXE:

(1) **Rating :**

This routine can be used to calculate overall heat transfer coefficient required for an exchanger for given end temperatures and surface area of heat exchange. The procedure uses piece wise integration over the entire surface of heat exchanger to calculate the overall heat transfer coefficient. This makes it possible without calculating the logarithmic mean temperature.

This program can also be used to calculate required surface area for an exchanger for given end temperatures and overall heat transfer coefficient.

The major assumption in above formulation is that the temperature changes only along the length of heat exchanger and not across it. In context of cross flow in a shell and tube heat exchanger usually with more than 10 baffles the assumption is appropriate.

This routine has two specialties, 1. It can handle variation in specific heat of fluids with temperature. 2. Usually number of tubes in different passes of tubes differ by 5 to 10 % and are not equal. This routine can handle different number of tubes in different passes.

The routine is limited only to the 2 tube pass or a U tube shell and tube exchanger.

(2) **Kern's Method :**

This routine is principally used in our calculation to arrive at a first design. The inputs are the fluid properties, tube parameters, and number of shell and tube passes. The complete formulation is given in chapter 2. This routine has its database contained in files named KERN.?00, where ? stands for positive number. File KERN.100 is containing data to convert BWG thickness of tube to thickness in inches. Files KERN.200 to KERN.700 contain data to get the shell diameter and tube count values for different tube diameters and tube patterns. Obviously the data is limited and can be added if available for more tube diameters and patterns.

(3) **Bell Method :**

The routine is available independently as well as through the Kern's Method.

For the purpose of rating of an existing heat exchanger it is necessary to call it independently from the main menu. It permits data input from a file as well as from screen. To begin data can be read from screen and when input from screen completes it makes a file of input data, BELLIN which can be used next time

if more runs are required.

For designing a heat exchanger it should be called from Kern's Method after executing Kern's method. This generates a preliminary design and this will then be passed to Bell method for further modifications.

The formulation of the method is given in chapter 2.

The method contains the data regarding the geometrical clearances of four type of heat exchangers for TEMA class R exchangers, so it can only be used for these four types of exchangers -

1 Fixed tube sheet type heat exchanger

2 U tube heat exchanger

3 Split backing ring type floating head heat exchanger

4 Pull through type heat exchangers.

The various types of heat exchangers are introduced in chapter 1 with their principal features which will help in their selection while running the program.

(4) Flow Induced Vibrations :

This unit is also available independently as well as through the Bell method.

When it is needed only to perform vibration check for an exchanger than this unit can be called independently, otherwise the call from Bell method will generate the vibration check list for the design under consideration.

The complete formulation is given in chapter 3. This unit checks for the possible damage by vortex shedding, fluid elastic whirling, and for the collision and baffle damage numbers. If

shell side fluid is a gas than this will also check for the acoustic vibrations additionally.

The file TSHEET.EXE can be used to calculate the full tube count. The actual tube count differs from the full tube count because of the reasons given in chapter 2, section 2.4.

The inputs are tube and tube sheet diameter, tube pitch and layout angle. The margin from outer edge can be changed. To enhance the creativity the coordinates for the tube to begin can also be altered. This feature might help in obtaining final tube sheet layout design for minimum leakage through the pass partition lanes in a heat exchanger with more than one tube passes.

This program is graphics implemented so the designer can get drawing of the actual tube layout, by deducting the number of tubes which has to be removed to provide pass partitions.

Using this software, drawings can be stored, viewed and compared together.

DESIGN PROBLEM AND RESULTS

This chapter describes the design problem considered to analyze. The known parameters of the design are as follows :

Hot stream consists of water to be cooled from 55 °C to 45 °C. The cold stream is cooling water available at 35 °C , which could be heated up to 40 °C. The mass flow rate of hot stream is 25 kg/sec. Length of tubes in exchanger is limited to 4 meter because of the limited space available for handling.

The thermodynamic properties of fluids used are as follows:

Specific heat for water on shell and tube side = 4186 J/kg.K

Thermal conductivity of water for shell and tube side = 0.62 W/mK

Dynamic viscosity of water on shell side = 8.1 E - 4 N.sec/m^2.

Dynamic viscosity of water on tube side = 9.1 E - 4 N.sec/m^2.

Specific gravities of water on both side = 1.

With the above information the first run of Kern's method has taken considering hot stream in shell side. The reason to assign shell side to the hot stream and the tube side to the cooling water is mentioned in chapter 1. The layout of tubes is taken triangular considering the hot water clean. The tube diameter 0.75 in.(19.05 mm) and pitch 0.9375 in. are selected from the available database of the program, because of the standardization reasons. The tube count and shell diameter is then calculated and taken from the nearest available standard

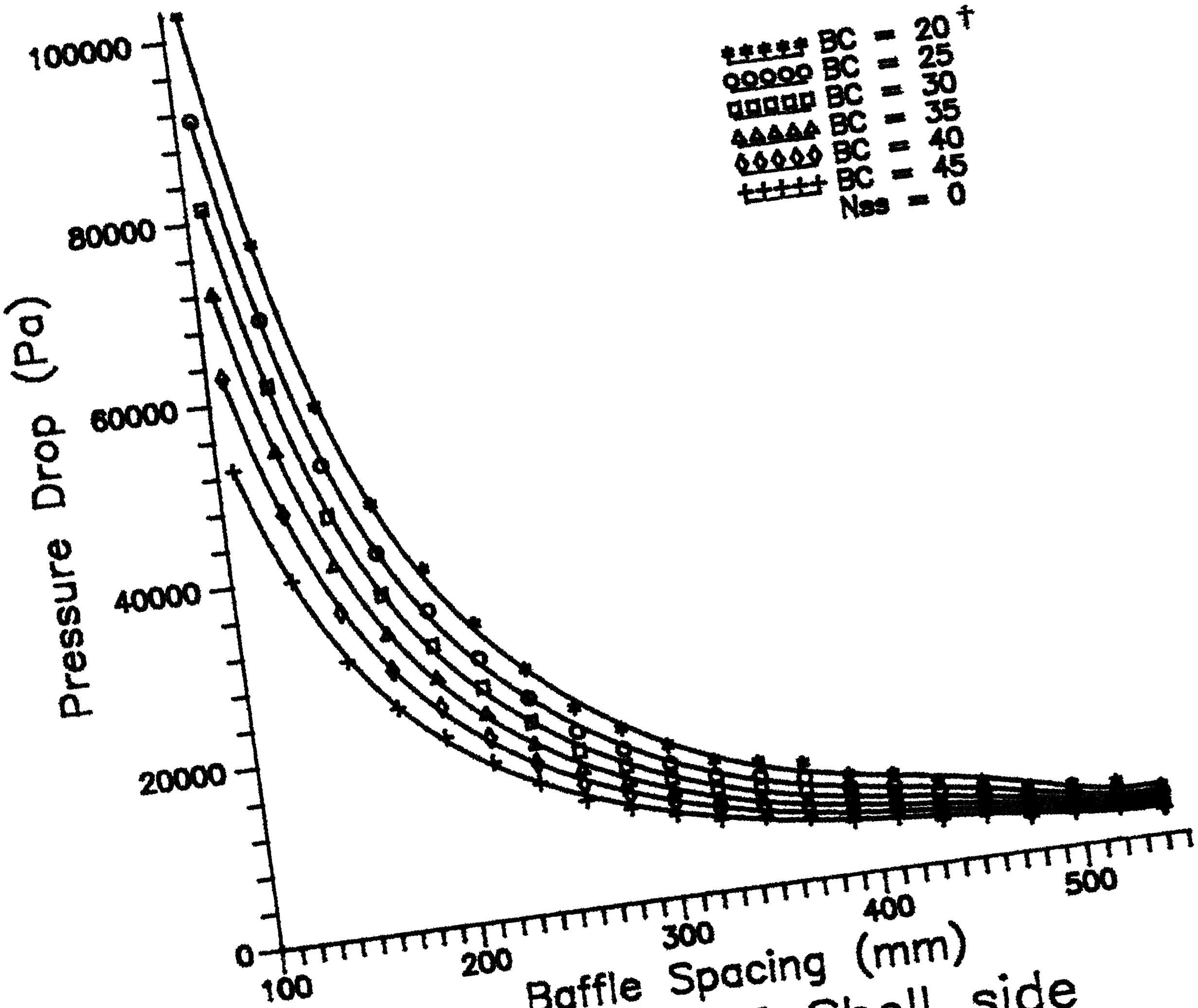

Fig. 8. Variation of Shell side Pressure drop with change in Baffle Spacing.

† BC is Baffle Cut (% of shell inner diameter)

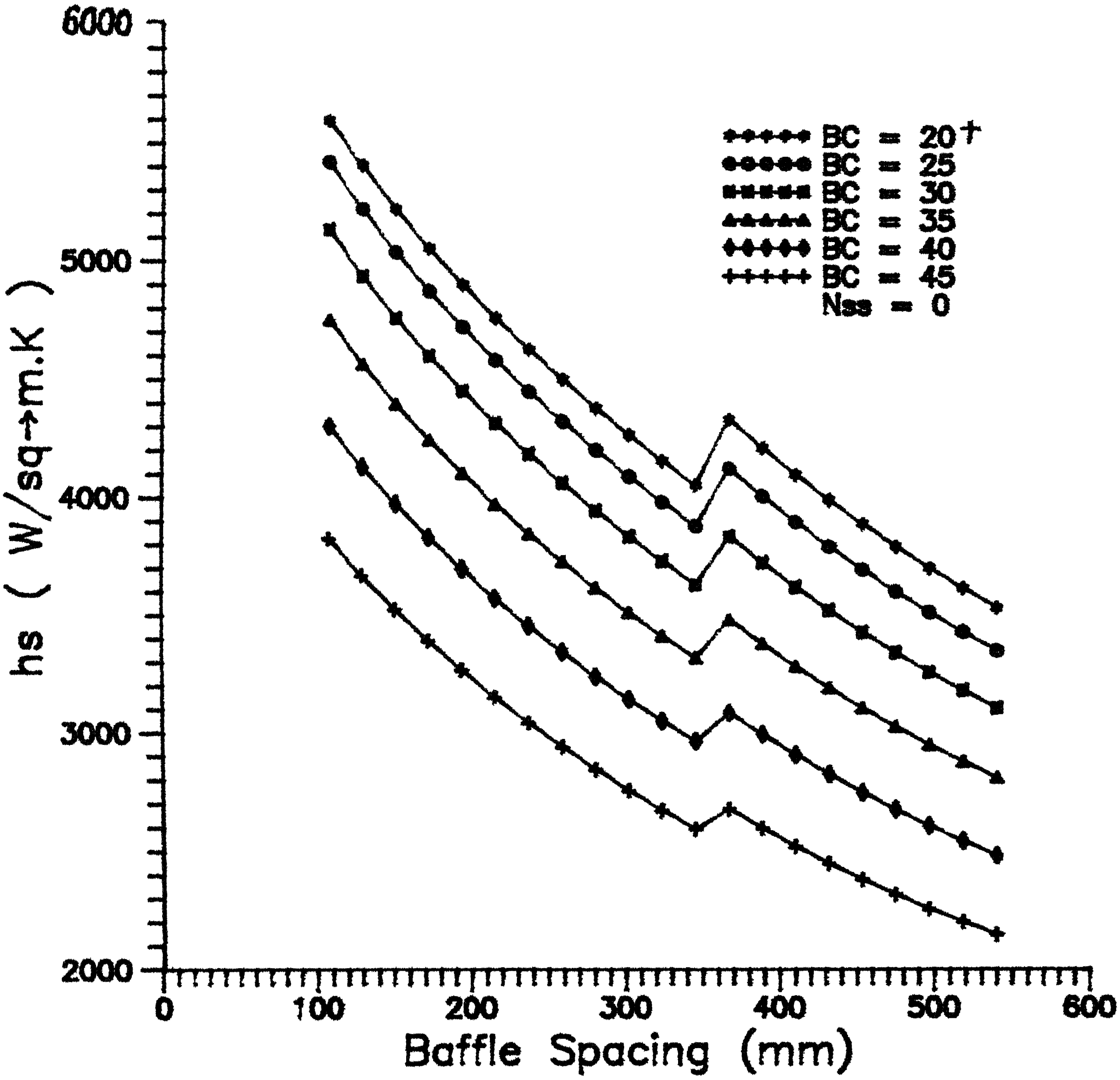

Fig. 9. Variation of Shell side Heat transfer coefficient with change in Baffle Spacing.

† BC is Baffle Cut (% of shell inner diameter)

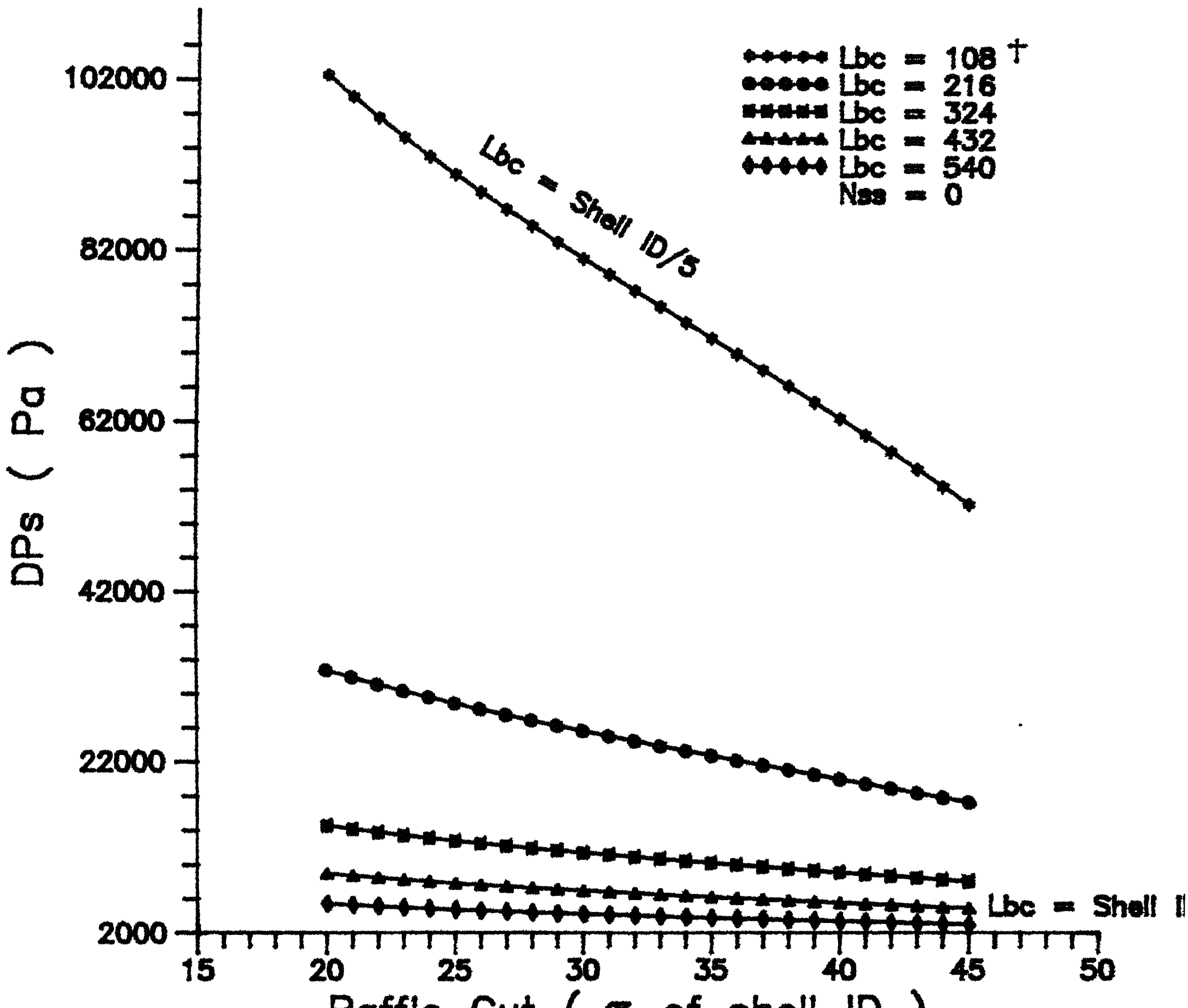

Fig. 10. Variation of Shell side pressure drop with change in baffle cut.

†Lbc is Central Baffle Spacing (mm)

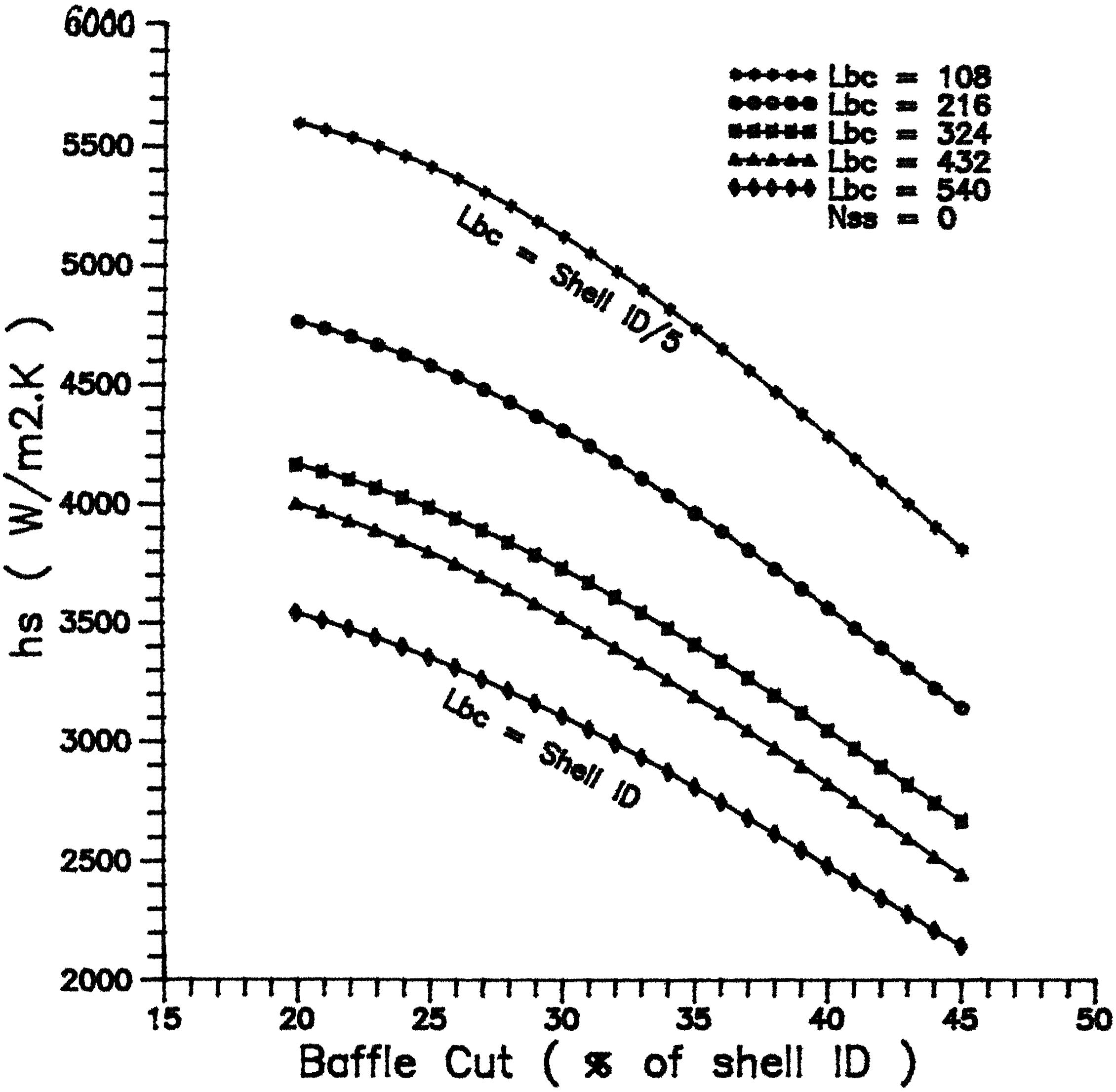

Fig. 11. Variation of Shell side pressure drop with change in baffle cut.

Lbc is Central Baffle Spacing (mm)

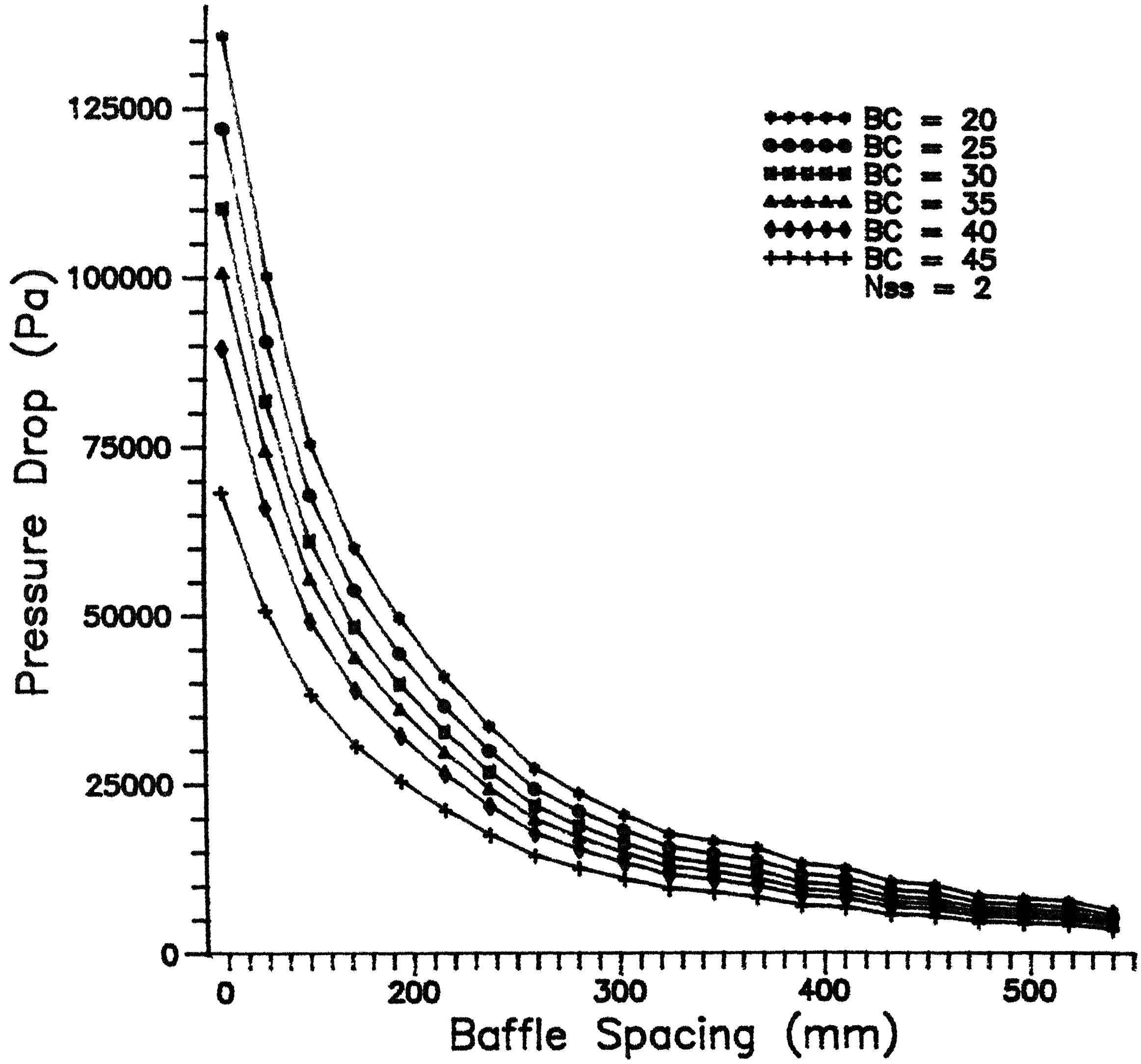

Fig. 12. Variation of Shell side Pressure drop with change in Baffle Spacing, for 2 pairs of sealing strips

BC is Baffle Cut (% of shell inner diameter)

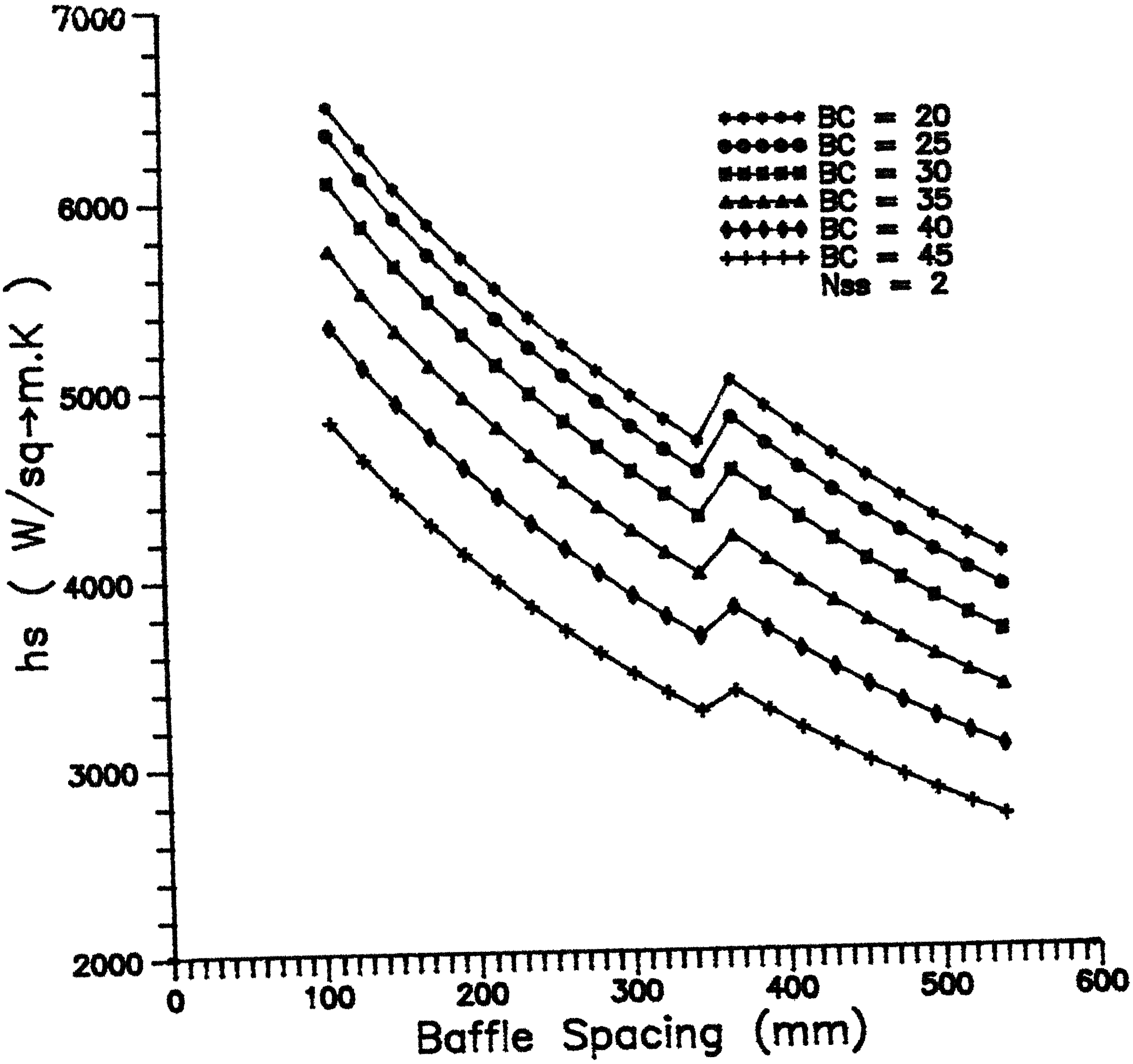

Fig. 13. Variation of Shell side h with change in Baffle spacing for 2 Sealing Strips.

BC is Baffle Cut (% of shell inner diameter)

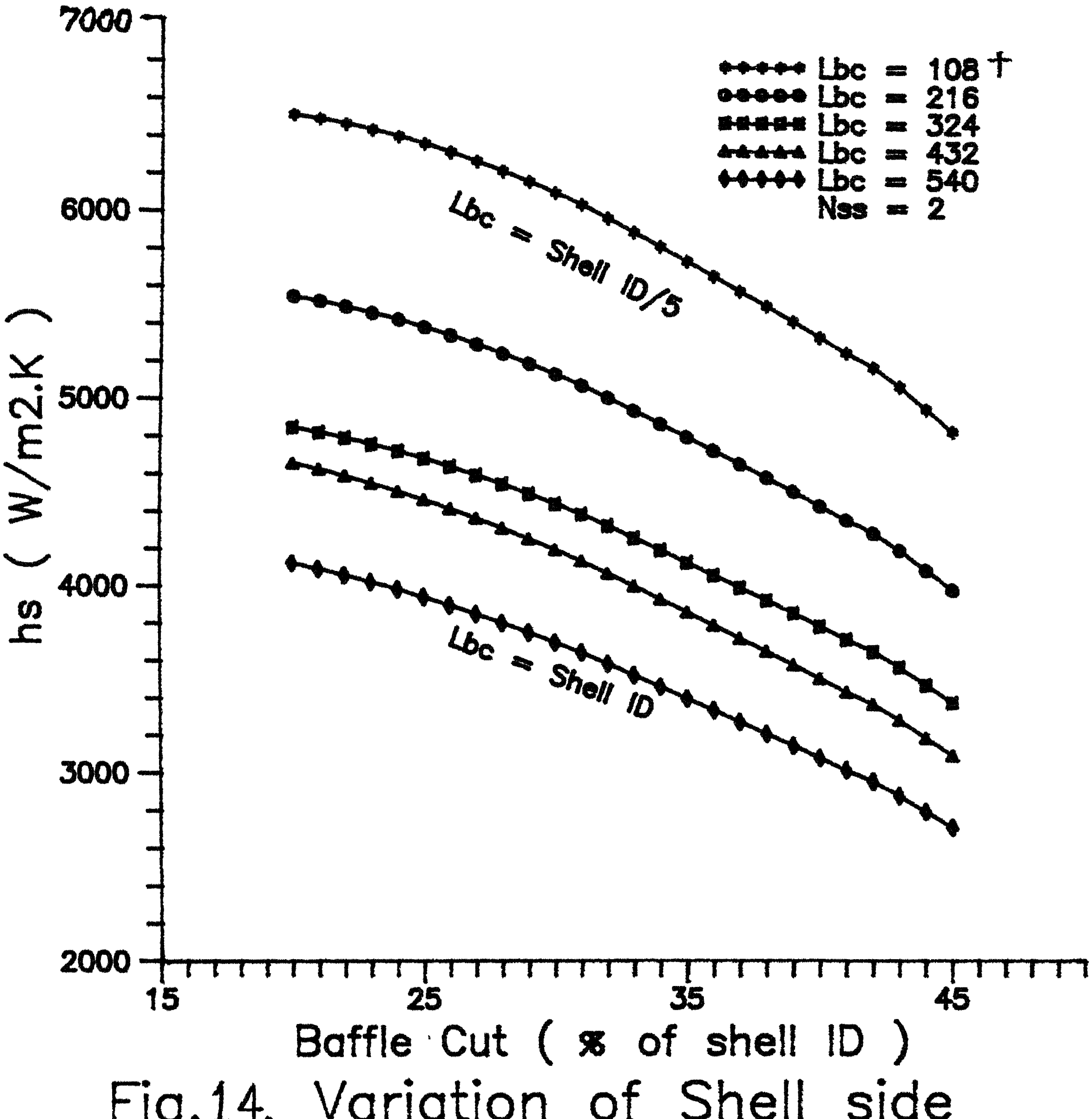

Fig.14. Variation of Shell side Heat transfer coefficient with change in Baffle Cut.

† Lbc is Central Baffle Spacing (mm)

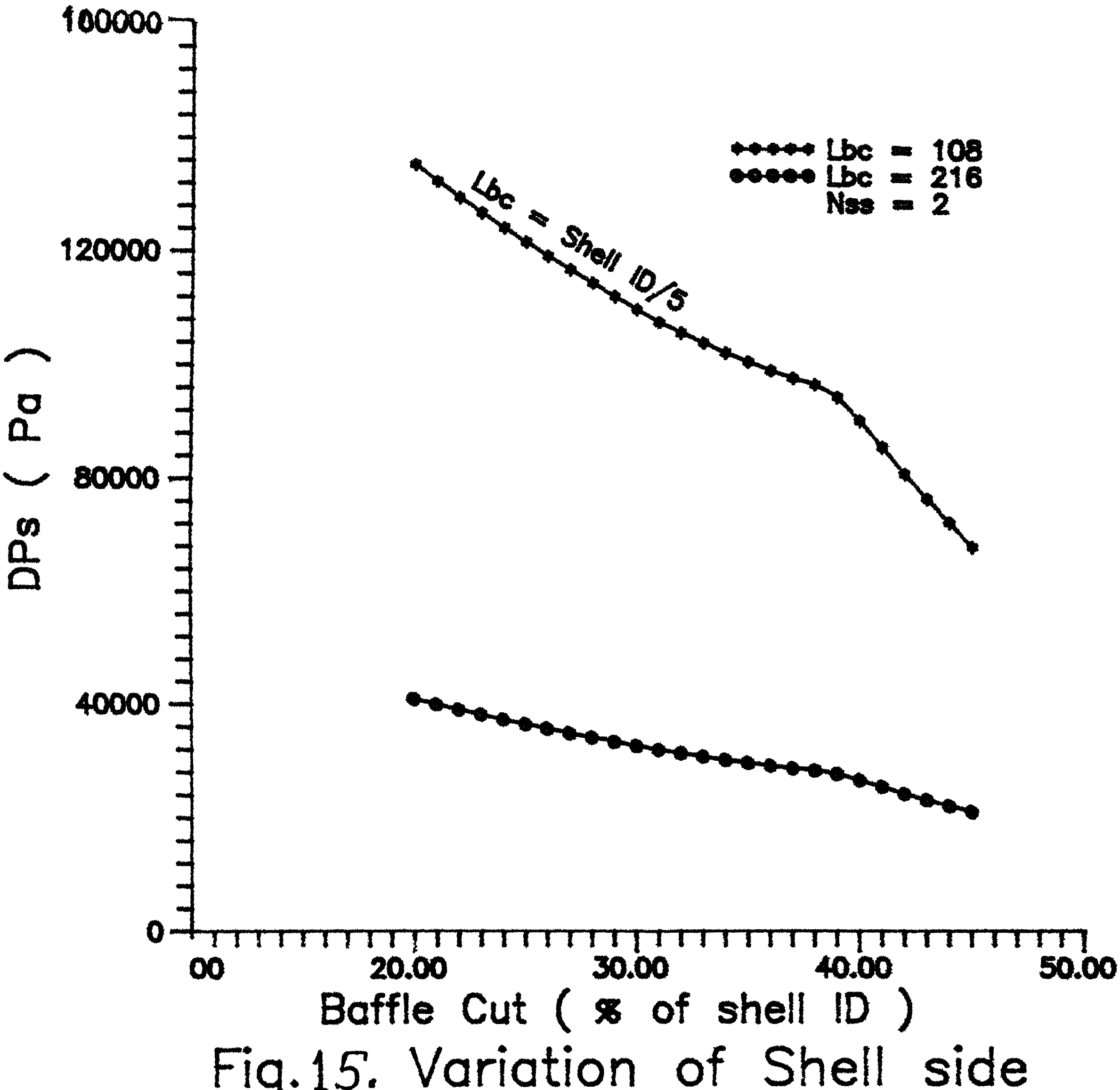

Fig.15. Variation of Shell side Pressure drop with change in Baffle cut, for 2 Sealing strips

Lbc is Central Baffle Spacing (mm)

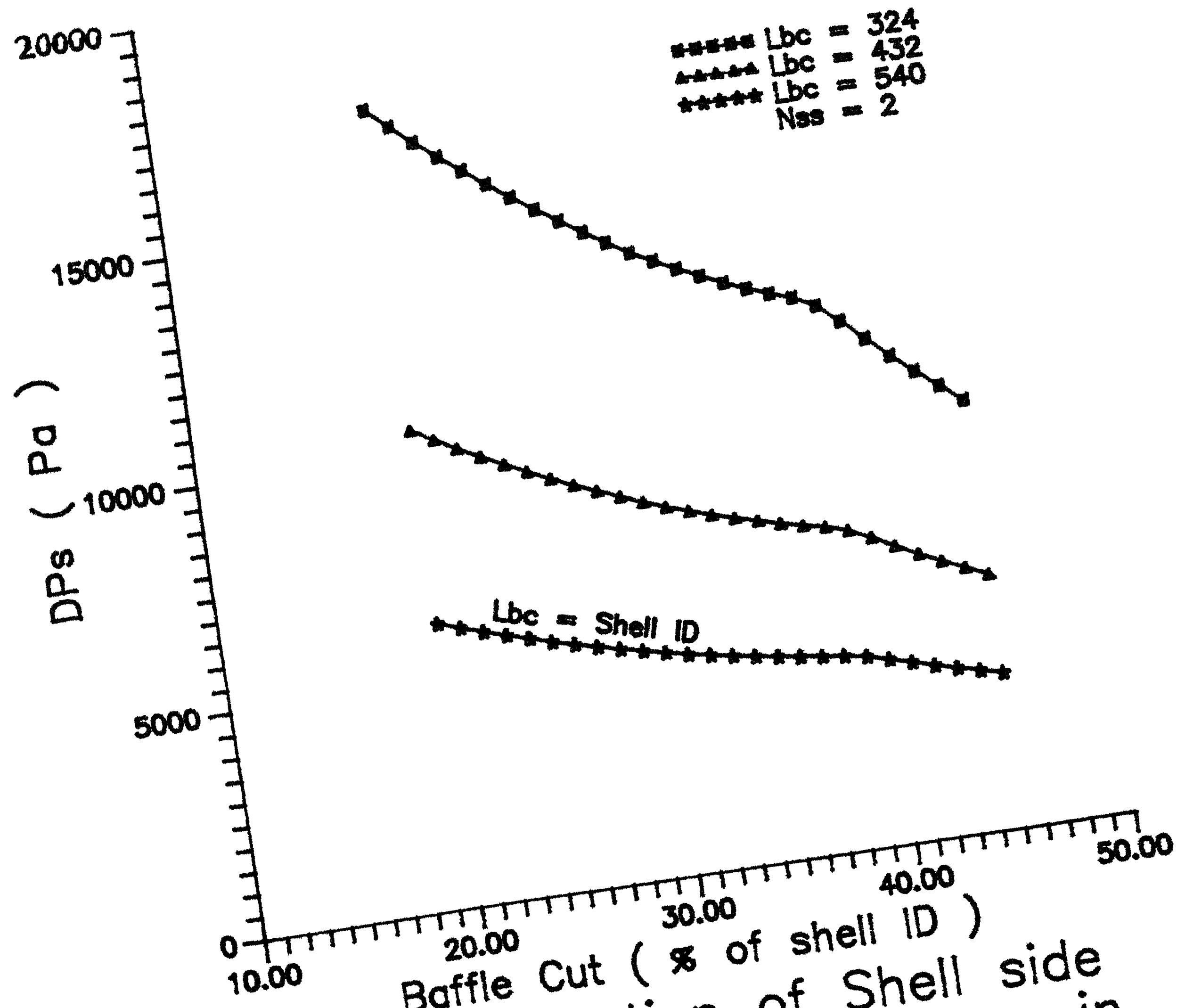

Fig. 16. Variation of Shell side Pressure drop with change in Baffle cut, for 2 Sealing strips

Lbc is Central Baffle Spacing (mm)

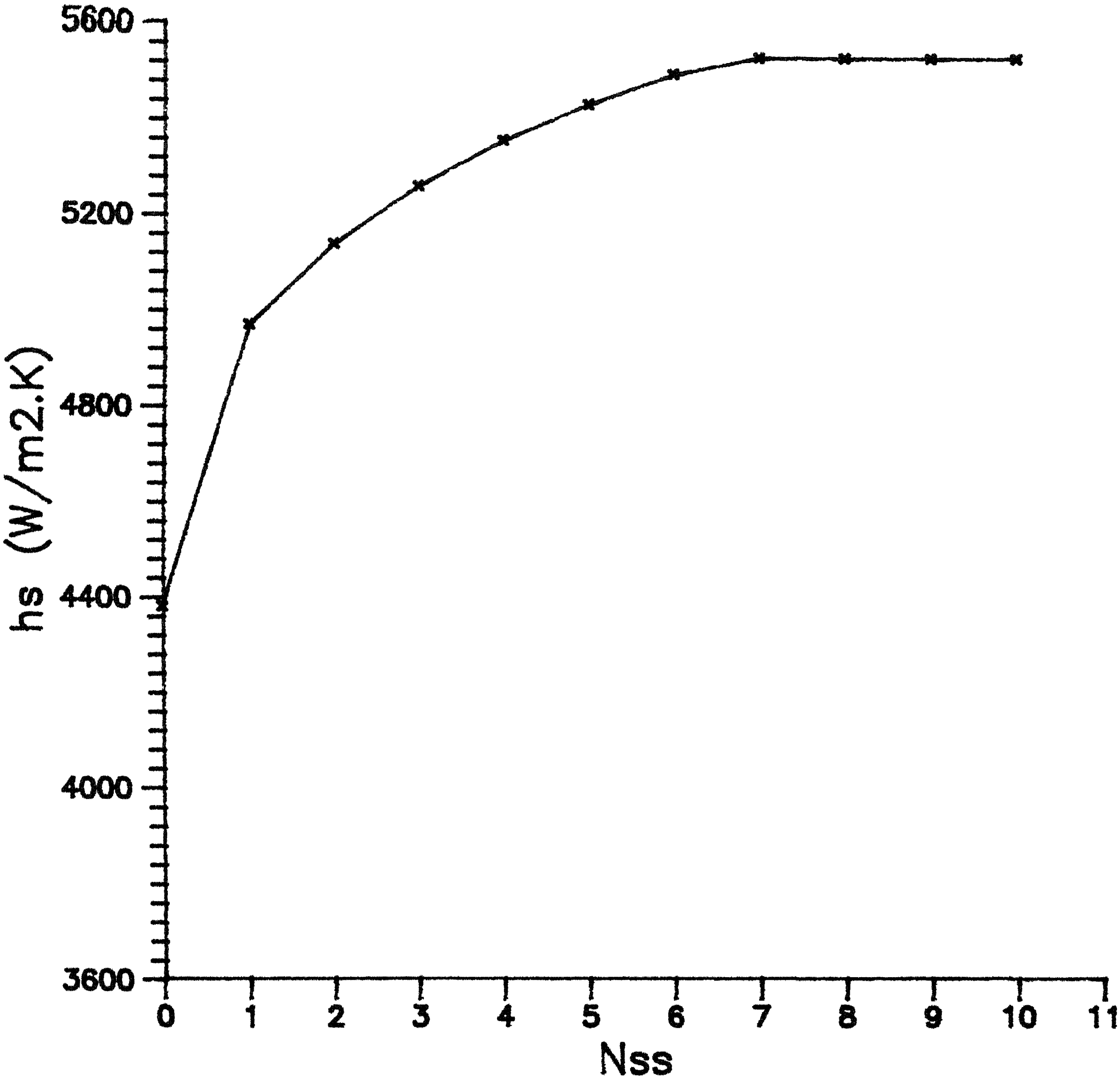

Fig. 17. Variation of Shell side hs with number of Pairs of Sealing strips for Bc=25 and Lbc=250mm.

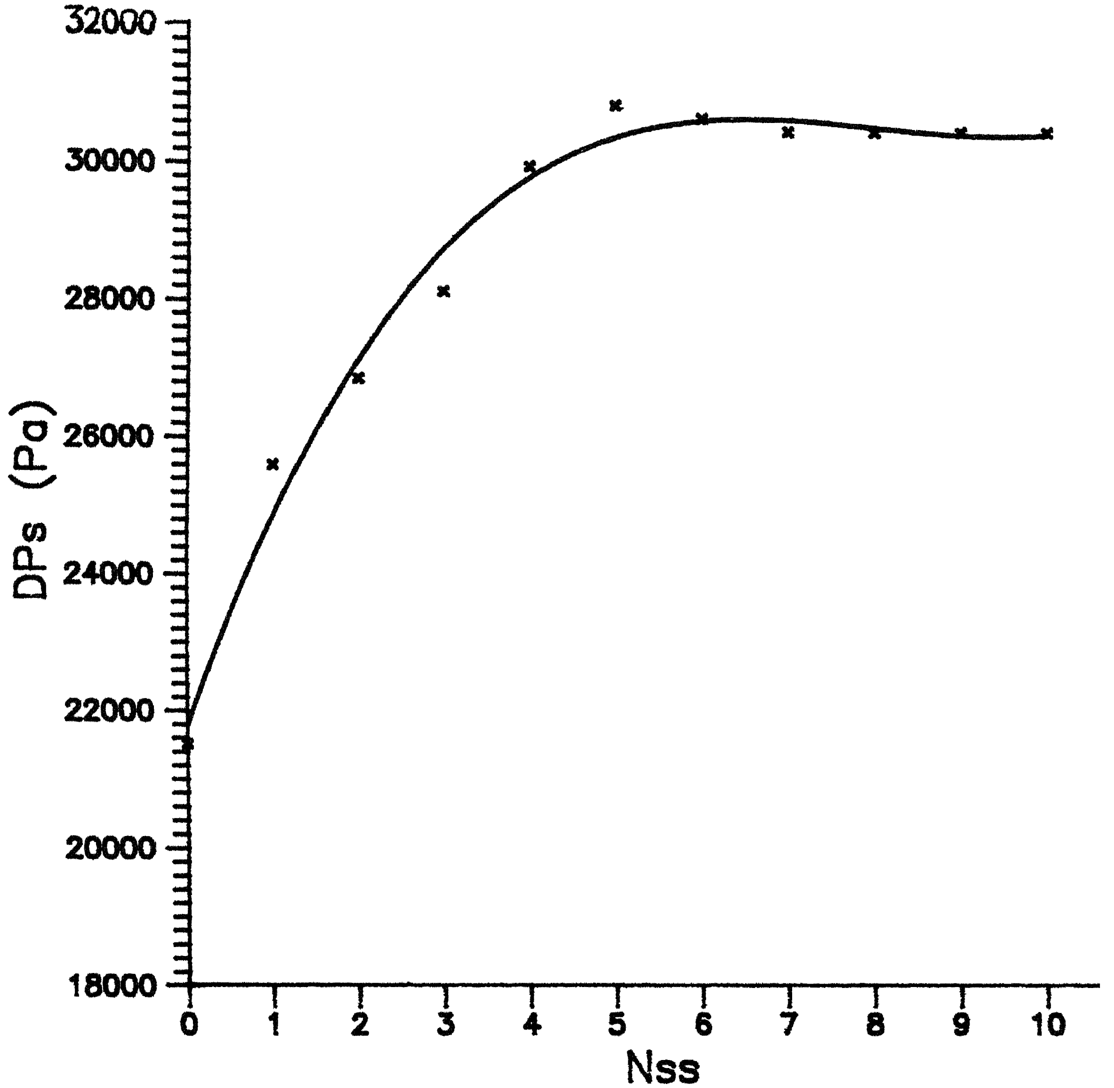

Fig. 18. Variation of Shell side Press
drop with Number of Pairs of Sea
strips for Bc=25 and Lbc=250mm

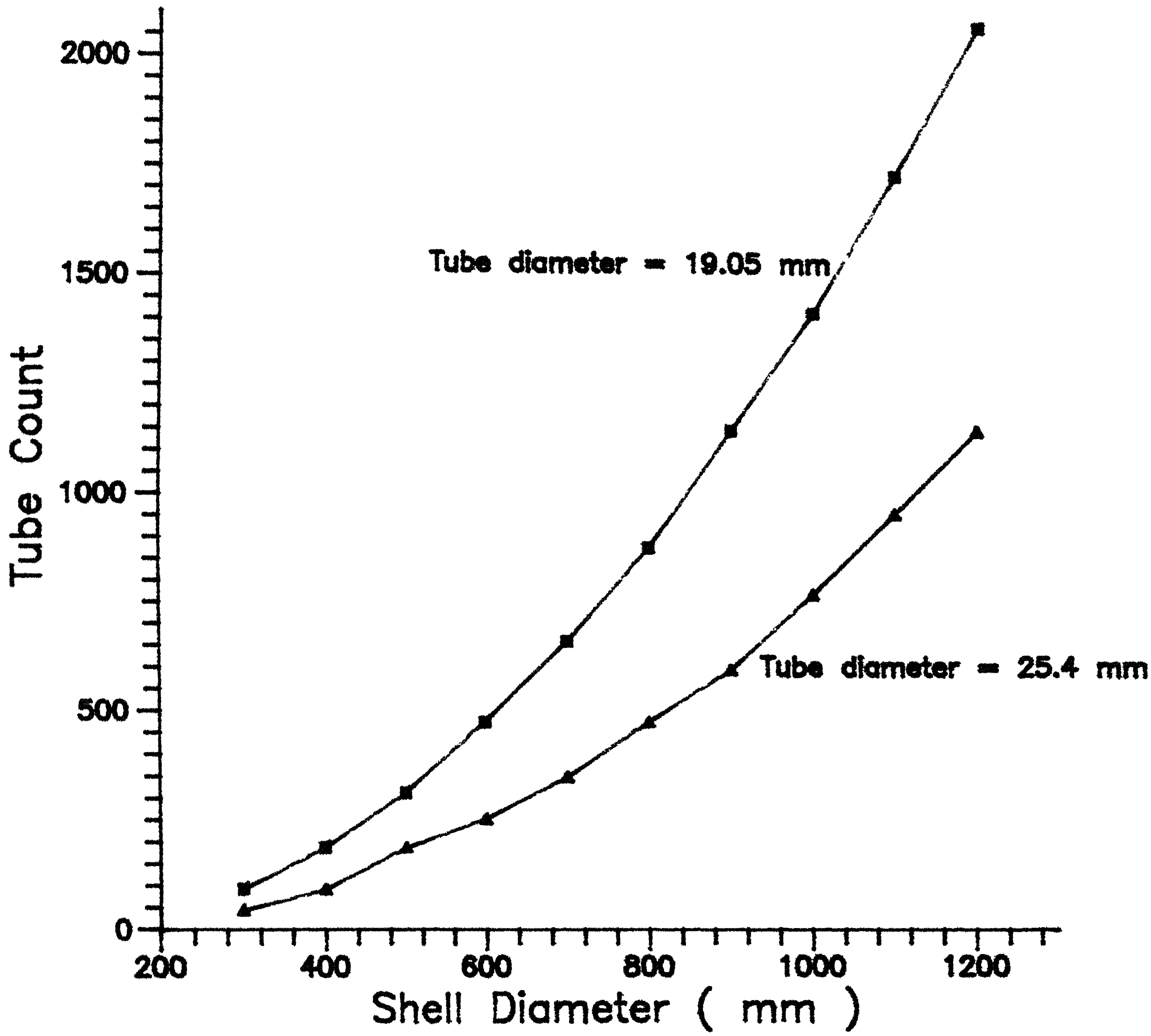

Fig. 19. Variation of Tube count with Shell diameter and Tube diameter for Pitch ratio 1.25, Layout angle 30 deg.

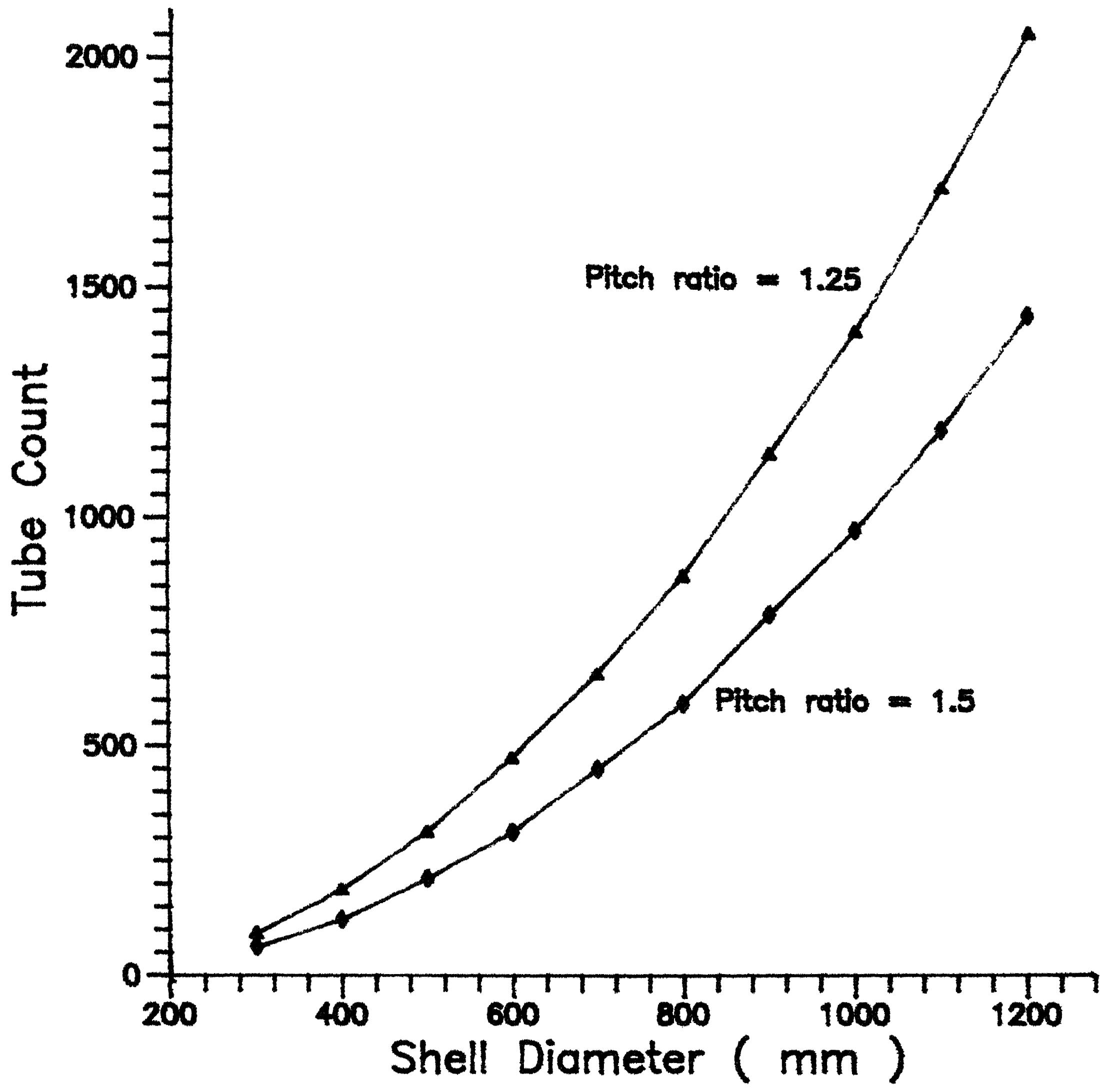

Fig. 20. Variation of Tube count with Shell diameter and Pitch ratio for Layout angle=30 deg., tube diameter=19.05 mm.

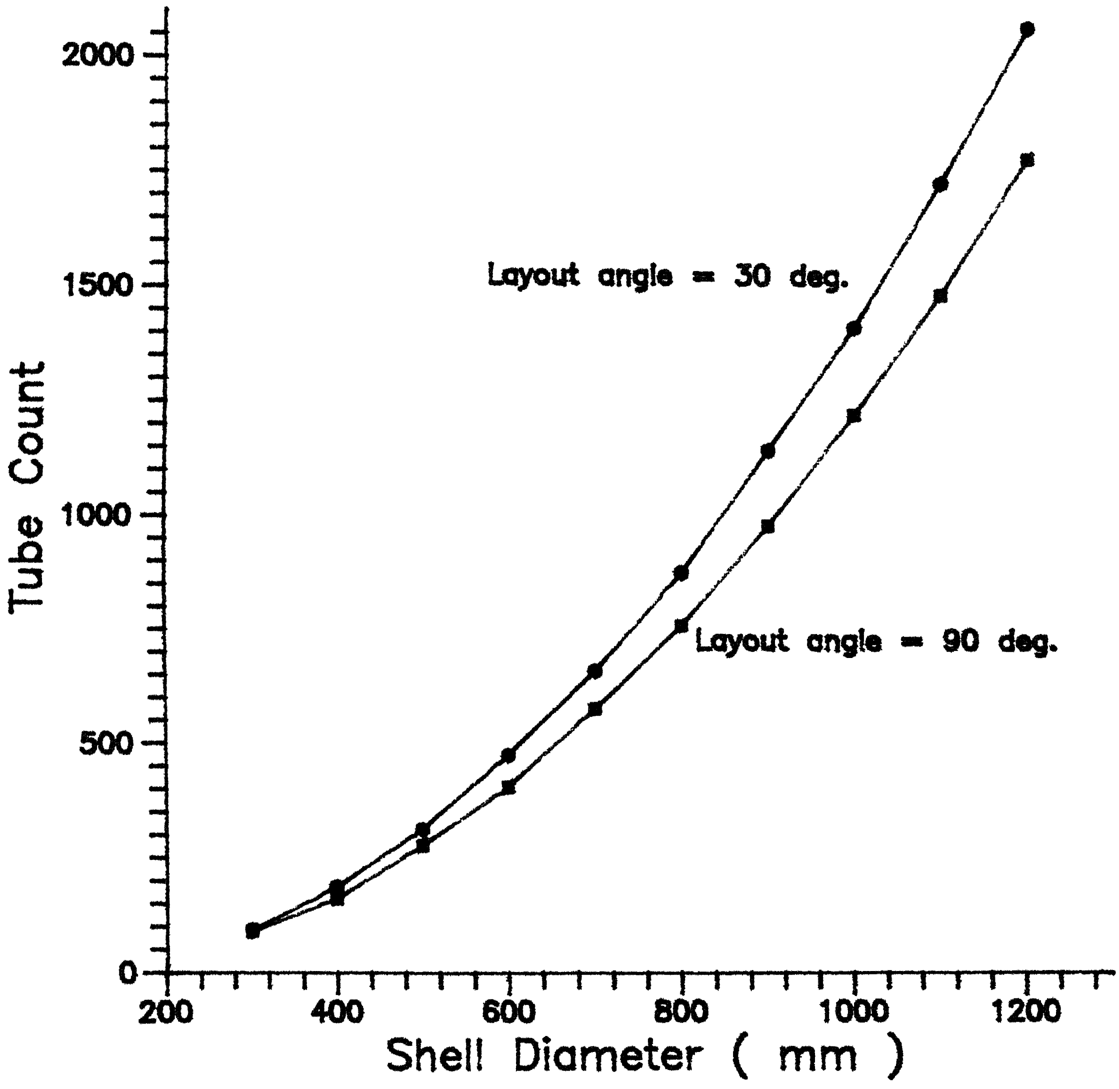

Fig. 2.1. Variation of Tube count with Shell diameter and Layout angle for pitch ratio = 1.25 and tube diameter = 19.05 mm.

diameter. Knowing shell diameter (540 mm) the baffle spacing is selected (250 mm), as suggested by TEMA[25] baffle spacing can be from one fifth of shell diameter to shell diameter. This completes the calculations of Kern method. The data calculated is then transferred to Bell method and thereafter to for the flow induced vibration routine.

The effect of varying the various shell side parameters on the shell side pressure drop and heat transfer coefficient is shown in plots (fig.8 to 21) for the heat exchanger under consideration. These plots can be used for the selection of baffle spacing , baffle cut and number of pairs of sealing strips for optimal heat transfer coefficient and pressure drop.

Selection of baffle spacing :

Fig.8. shows that the shell side pressure drop is decreasing exponentially for increase in baffle spacing from one fifth of shell diameter to that equal to shell diameter. The plot shows that decrease in shell side pressure drop is not significant as the baffle spacing is increased from 250 mm.

The fig.9. shows the corresponding decrease in shell side heat transfer coefficient by increasing baffle spacing. The diagram shows a sudden rise in heat transfer coefficient for baffle spacing about 365 mm. The reason is that after a baffle spacing of 365 mm, according to TEMA recommendations the baffle hole to tube diameter clearance is reduced to 1/64" from 1/32". This results in reduction of leakage stream A (fig.4.). In our computation the value of factor J_L increases from about 0.72 to

0.81, hence the sudden rise in shell side heat transfer coefficient.

Both of the above discussed fig.8 and fig.9. are plotted considering there is no sealing strips , i.e. $N_{ss} = 0$.

Selection of Baffle Cut :

Fig.10. shows the decrease in shell side pressure drop with increase in baffle cut. The drop is proportional.

Fig.11. shows the variation of shell side heat transfer coefficient for increase in baffle cut. The graph shows that drop in shell side heat transfer coefficient is more steep as baffle cut is increased after 25 % of shell diameter. The reason is that for small baffle cut (with small baffle spacing) the amount of leakage streams is higher and as baffle cut and spacing are increased the leakage streams also reduces because of reduction in flow resistance. As baffle cut and spacing are further increased the amount of parallel flow increases which also reduces the shell side heat transfer coefficient. Thus the baffle cut of about 25 % of shell diameter seems appropriate particularly for small baffle spacings. For larger baffle spacings 20% baffle cut can be used.

Selection of Number of pairs of sealing strips :

Fig.12. shows the drop in shell side pressure drop with increase in baffle spacing considering two pair of sealing strips. Similarly fig.13 and fig.14 shows variation of shell side heat transfer coefficient for increase in baffle spacing and baffle cut respectively considering two pairs of sealing strips. The trend is almost same as it was without sealing strips in fig.8 to fig.11. The fig.15 shows the decrease in pressure drop

with increase in baffle cut, As expected the decrease is steep for smaller baffle spacings (fig.15), than for the larger baffle spacings(fig.16). The curve shows a comparatively steep fall in pressure drop for baffle cut more than 40 %. For such larger cuts the shell side flow essentially becomes parallel that is why the drop in pressure drop. The shell side heat transfer coefficient also suffers a steep fall for large baffle cuts because of same reason (fig.14).

Fig.17. shows the variation of shell side heat transfer coefficient for increase in sealing strips. As sealing strips are provided to reduce the amount of bundle bypass streams. The curve shows that the rise in heat transfer coefficient is insignificant for more than 5 pairs of sealing strips, and not as steep for more than 2 pairs of sealing strips.

Fig.18.shows the rise in shell side pressure drop for increase in sealing strips, the trend is similar to that of the heat transfer coefficient (fig.17).

Designing for compactness :

Fig.19 to 21 shows the variation of full tube count for variation in shell diameter for different tube diameter, pitch ratio and layout angle respectively. These curve are produced by data generated from Interactive Graphics Program TSHEET. These curves depicts that the compactness (larger tube count), is affected maximum by the change in tube diameter and change in layout angle affects minimum. So as far as the selection of physical parameters is concerned the change in layout angle must be tried first, change in pitch ratio next and change in tube

diameter the last.

The output specifications of the designed heat exchanger is given in Appendix C, as is obtained from the computer. It contains output from the three routines , The Kern, Bell, and finally the vibration check report from Vibunit.

CONCLUSIONS AND SCOPE

An interactive, menu driven code for thermal hydraulic design and rating of shell and tube heat exchangers has been developed. The application to the typical design problem produced results were logically discussed in previous chapter. Considering the importance of flow induced vibration checking, right at the time of thermal hydraulic design, the routine has been added, which is also accessible independently from rest of the code.

The program is suitable only for single phase exchangers, with plain tube only. The liquid metal flow also not considered.

Inclusion of correlations and data for low fin tubes and two phase flow as well as for liquid metal heat transfer can further enhance the utility of this software.

Another improvement could be addition of optimizing routines to let computer make decisions about optimal parameters like flow velocity, cooling water temperatures, and area for heat transfer etc. Yet another possibility is of interfacing mechanical design and cost analysis codes and globally optimizing the design.

Ref [4] and [5] also suggest additional features to supplement a design code.

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APPENDIX A

AUXILIARY CALCULATIONS

(a) Segmental Baffle Window Calculations

The central angles of the baffle cut with the inside of shell, θ_{ds} and the angle intersecting the tube diameter

$$D_{ctl} = D_{otl} - D_t \text{ are} \qquad (\text{ see fig.22 }) \qquad \ldots\ldots(A.1)$$

$$\theta_{ds} = 2\cos^{-1}(1-2B_c/100) \qquad \ldots\ldots\ldots(A.2)$$

$$\theta_{ctl} = 2\cos^{-1}\left[\frac{D_s}{D_{ctl}}\left(1-2\frac{B_c}{100}\right)\right] \qquad \ldots\ldots\ldots(A.3)$$

where B_c is baffle cut expressed as percentage of shell inner diameter. D_s is shell inner diameter, D_{otl} is outer tube limit as shown geometrically in fig.22.

If baffles are of window type,(generally used in floating head type exchangers) then area between D_s and D_{otl} is blocked, so the angle is referred to D_{otl},

$$\theta_{otl} = 2\cos^{-1}\left[\frac{D_s}{D_{otl}}\left(1-2\frac{B_c}{100}\right)\right] \qquad \ldots\ldots\ldots(A.4)$$

(b) Baffle Window Flow Areas

The gross window flow area, *i.e.* without tubes is-

$$A_{wg} = \frac{\pi}{4} D_s^2\left[\frac{\theta_{ds}}{2\pi} - \frac{\sin\theta_{ds}}{2\pi}\right] \qquad \ldots\ldots\ldots(A.5)$$

If window baffles are used then θ_{ds} in above equation is replaced by θ_{otl}. Assuming a uniform tube field the fraction of tubes in one window, F_w and the fraction in cross flow between baffle tips F_c are

$$F_w = \frac{\theta_{ctl}}{2\pi} - \frac{\sin\theta_{ctl}}{2\pi} \qquad \ldots\ldots\ldots(A.6)$$

$$F_c = 1-2F_w \qquad \ldots\ldots\ldots(A.7)$$

If tube field is not uniform because of pass lanes or tubes removed in the nozzle entry zone then this correction has to be

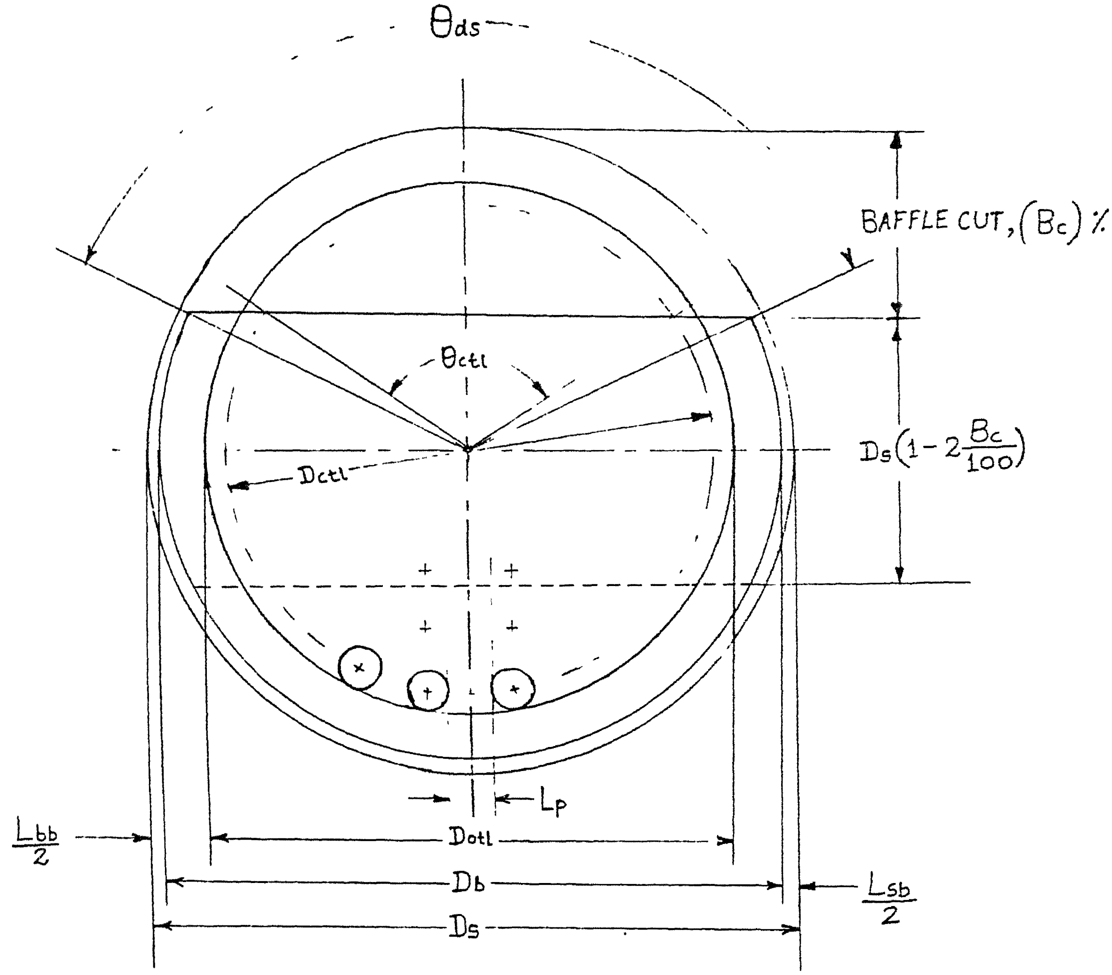

Fig.22. BAFFLE GEOMETRY

separately calculated.

The baffle window area occupied by tubes, A_{wt} is

$$A_{wt} = N_t\, F_w\, \frac{\pi D_t^2}{4} \qquad \text{.......(A.8)}$$

and the net flow area through one window, A_w is

$$A_w = A_{wg} - A_{wt} \qquad \text{.......(A.9)}$$

where N_t is total number of tubes in exchanger,

D_t is tube outside diameter.

(c) Equivalent Hydraulic Diameter for the Window, D_w

This value is required only for pressure drop calculation in laminar region (R_e < 100).

$$D_w = \frac{4\, A_w}{\pi\, D_t\, F_w\, N_t + \pi\, D_s(\theta_{ds}/2\pi)} \qquad \text{.......(A.10)}$$

neglecting the baffle edge thickness.

(d) Number of Effective Tube Rows in Cross Flow, N_{tcc} The number of tube rows is a function of tube layout and pitch parameters. L_{pp} is defined in fig.23. For one cross flow section between baffle tips

$$N_{tcc} = \frac{D_s}{L_{pp}} \left(1 - \frac{B_c}{100} \right) \qquad \text{.......(A.11)}$$

(e) Number of Effective Tube Rows in Baffle Window, N_{tcw}

It is based on interpretation of the window flow pattern. data from Delaware method can be approximated as 0.4 times the tube field in the window. This distance is crossed twice in each window, hence

$$N_{tcw} = \frac{0.8}{L_{pp}}\, D_s \left[\frac{B_c}{100} - \frac{D_s - D_{ctl}}{2D_s} \right] \qquad \text{.......(A.12)}$$

(f) Number of Baffles, N_b

For calculation purpose number of baffles is determined from tube length and central baffle spacing even if larger end

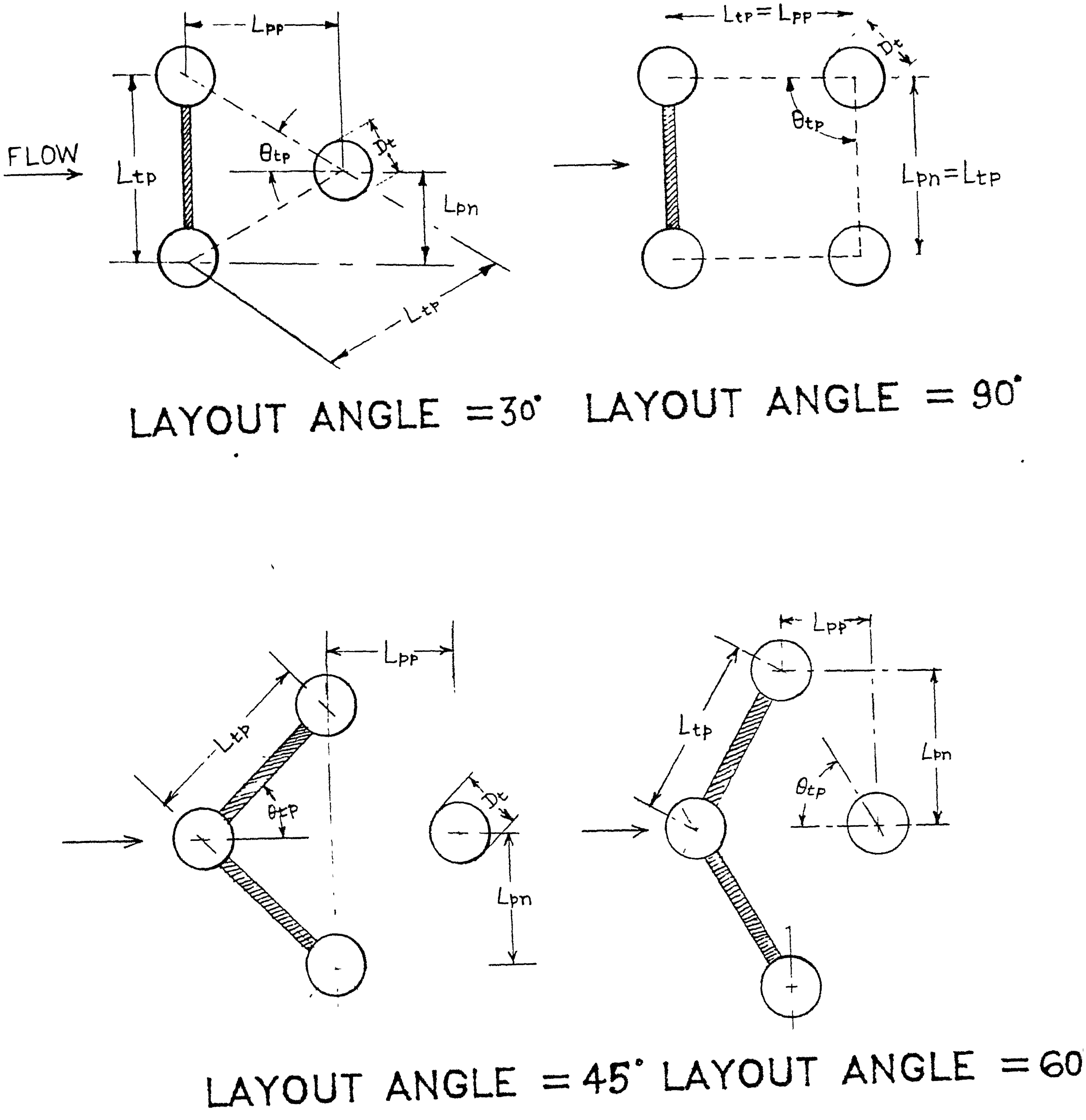

Fig.23. TUBE LAYOUT PARAMETERS

spacings are used. The tube length L_{ti} is defined in fig.5B.

$$N_b = \frac{L_{ti}}{L_{bc}} - 1 \qquad \text{.......(A.13)}$$

(g) Bundle Shell Bypass

The bypass area within one baffle A_{ba}, is

$$A_{ba} = L_{bc}[(D_s - D_{otl}) + L_{pl}] \qquad \text{.......(A.14)}$$

where L_{pl} is the tube lane partition bypass width as follows:

$$L_{pl} = \begin{cases} \text{0 for all standard calculations.} \\ \text{1/2 of real dimension of tube lane partition, } L_p. \\ \text{For estimation purpose } L_p = D_t. \end{cases}$$

(h) Shell-Baffle Leakage Area, A_{sb}

The diametral clearance between the shell diameter D_s and the baffle diameter D_b is designated L_{sb}. The shell baffle leakage area within the circle segment of the baffle is

$$A_{sb} = \pi D_s \frac{L_s}{2} \frac{2\pi - \theta_{ds}}{2\pi} \qquad \text{.......(A.15)}$$

where $L_{sb} = D_s - D_b$

(i) Tube Baffle Leakage Area, A_{tb}

The diametral clearance L_{tb} is determined from specified manufacturing tolerances, which are given in TEMA. The total leakage area for a baffle is

$$A_{tb} = \frac{\pi}{4}[(D_t + L_{tb})^2 - D_t^2]N_t(1 - F_w) \qquad \text{.......(A.16)}$$

(j) Minimum Bundle Cross Flow Area, A_{mb}

This is the minimum cross flow area for one baffle at the shell center line determined as follows:

$$A_{mb} = L_{bc}\left[L_{bb} + \frac{D_{ctl}}{(L_{tb})_{eff}}(L_{tp} - D_t)\right] \qquad \text{....... (A .17)}$$

where L_{bc} is the central baffle spacing, $(L_{tb})_{eff}$ is L_{tp} for 30 and 90 degree layouts , 0.707L_{tp} for 45 deg. layout and 0.866L_{tp} for 60 degree layout,(shown in fig.23.).

APPENDIX B

This appendix lists the polynomials used by routine BELLUNIT , *the terms used are defined in chapter 2.* The polynomials are fitted by best fit method for specified range only. Function pow(x,n) is used to represent x^n.

Friction Factor f and Heat transfer factor j_h for shell side Calculation:

For layout angle = 30 deg

Res >= 10 and Res <= 300 log(j):=2.5727e-1+3.6252e-1*log(Res)

Res > 300 and Res <= 1e6 log(j):=-1.3483+6.3791e-1*log(Res)

for pitch ratio = 1.25

Res >=10 and Res <=300 log(f) = 1.9311 - 1.5163 * log(Res) + 8.0455e-2 * sqr(log(Res))

Res >300 and Res <= 1e4 log(f):=2.977 - 1.0019 * log(Res) + 5.0484e-2 * sqr(log(Res))

Res > 1e4 and Res <= 1e5 log(f) = -2.2246 + 2.8706e-1 * log(Res) -2.8118e- 2*sqr(log(Res))

Res > 1e5 and Res <= 1e6 log(f) = 3.7237 - 8.8957e-1 * log(Res) + 2.9240e-2 * sqr(log(Res))

For pitch ratio = 1.33

Res >= 10 and Res <= 300 log(f) = 4.6744 - 1.5446 * log(Res) + 9.1024e -2 * sqr(log(Res))

Res > 300 and Res <= 1e4 log(f) = 2.4338 - 8.9004e-1 * log(Res) + 4.3886e-2 * sqr(log(Res))

Res > 1e4 and Res <= 1e5 log(f) = 1.2393 - 3.9821e-1 * log(Res) + 4.5795e-3 * sqr(log(Res))

Res > 1e5 and Res <= 1e6 log(f) = 1.3153e1- 2.444 * log(Res) +

9.2416e-2 * sqr(log(Res))

For pitch ratio = 1.5

Res >= 10 and Res <= 300 log(f) = 3.534 - 1.3729 * log(Res) +8.7588e- 2 * sqr(log(Res))

Res > 300 and Res <= 1e4 log(f) = 2.9054 - 1.0789 * log(Res) + 5.6273e-2 *sqr(log(Res))

Res > 1e4 and Res <= 1e5 log(f) = -9.0367e-1 - 1.5826e-2 * log(Res) - 1.4204e-2 * sqr (log(Res))

Res > 1e5 and Res <= 1e6 log(f) = -2.367e-1 - 3.6424e-1 * log(Res) + 1.0979e-2 *sqr (log(Res))

For layout angle = 45

Res >= 10 and Res <= 300 ln(j) = 2.8378e-1 + 3.5609e-1 * log(Res)

Res > 300 and Res <= 1e6 ln(j) = -1.4142 + 6.4911e-1 * log(Res)

for pitch ratio = 1.25

Res >= 10 and Res <= 300 log(f) = 3.9259 - 9.4360e-1 * log(Res)

Res > 300 and Res <= 1e6 log(f) = -8.754e-1 + 1.2664e-1 * log(Res) -5.3203e-2 *sqr (log(Res)) + 2.3701e-3 * pow(log(Res),3)

for pitch ratio = 1.33

Res >= 10 and Res <= 300 log(f) = 4.0674 - 1.2915 * log(Res) + 5.498e-2 * sqr(log(Res))

Res > 300 and Res <= 1e6 log(f) = -4.4619e-1 - 6.03e-2 * log(Res) - 3.2313e-2 * sqr (log(Res)) + 1.6765e-3 * pow(log(Res),3)

for pitch ratio = 1.5

Res >= 10 and Res <= 300 log(f) = 3.4945 - 1.3859 * log(Res) + 8.2815e-2 * sqr(log(Res))

Res > 300 and Res <= 1e6 log(f) = -1.7959 + 2.4112e-1 * log(Res) - 5.3789e-2 * sqr(log(Res)) + 2.1602e-3 * pow(log(Res),3)

For layout angle = 90

Res >= 10 and Res <= 300 ln(j) = -3.064e-1 + 4.2553e-1 * log(Res)

Res > 300 and Res <= 1e6 ln(j) = -1.6355 + 6.5961e-1 * log(Res)

for pitch ratio = 1.25

Res >= 10 and Res <= 300 log(f) = 4.0516 - 9.9106e-1 * log(Res)

Res > 300 and Res <= 2000 log(f) = 5.6412e1 - 2.3891e1 * log(Res)

+ 3.2345 * sqr(log(Res)) -1.454e-1 * pow(log(Res),3)

Res > 2000 and Res <= 2e4 log(f) = 1.0152e1 - 4.882 * log(Res) +

6.3149e-1 * sqr(log(Res)) -2.6622e-2*pow(log(Res),3)

Res > 2e4 and Res <= 1e6 log(f) = -1.2899e1 + 3.3987 * log(Res) -

3.3493 e-1 * sqr(log(Res)) + 1.0271e-2 * pow(log(Res),3)

for pitch ratio = 1.33

Res >= 10 and Res <= 300 log(f) = 3.5777 - 9.8343e-1 * log(Res)

Res > 300 and Res <= 2000 log(f) = 3.9046e1 - 1.735e1 * log(Res) +

2.4138 * sqr (log(Res)) - 1.113e-1 * pow(log(Res),3)

Res > 2000 and Res <= 2e4 log(f) = -1.5003e-1 - 1.3218 * log(Res)

+ 2.2156e-1 * sqr(log(Res)) - 1.1061e-2 * pow(log(Res),3)

Res > 2e4 and Res <= 1e6 log(f) = 5.6469 - 1.2588 * log(Res) +

4.6627e-2 * sqr (log(Res))

for pitch ratio = 1.5

Res >= 10 and Res <= 300 log(f) = 2.8302 - 9.4005e-1 * log(Res)

Res > 300 and Res <= 2000 log(f) = 4.0023e1 - 1.8296e1 * log(Res)

+ 2.5919 * sqr(log(Res)) - 1.2107e-1 * pow(log(Res),3)

Res > 2000 and Res <= 2e4 log(f) = -9.8516 + 1.6407 * log(Res) -

7.9541e-2 * sqr(log(Res)) - 1.1427e-3 * pow(log(Res),3)

Res > 2e4 and Res <= 1e6 then log(f) = 4.1080-1.0597 * log(Res) +

3.9501e-2 * sqr(log(Res))

f = exp(log(f))

jh = exp(log(j))

j = jh/Res

Factors J_L, J_b, R_l, R_{bf}, and J_c for shell side heat transfer coefficient determination

for rlm >= 0.0 and rlm <= 0.8

for rs = 0

JL = 9.917e-1 - 1.1170*rlm + 9.2394e-1 *rlm*rlm - 2.9242e-1 * pow(rlm,3)

for rs = 0.25

JL = 1.0081 - 1.4913*rlm + 1.4881*rlm*rlm - 6.2222e-1 * pow(rlm,3)

for rs = 0.5

JL = 9.9736e-1 - 1.6064*rlm + 1.431*rlm*rlm - 5.2778e-1 * pow(rlm,3)

for rs = 0.75

JL = 1.0166 - 1.9441*rlm + 1.7313*rlm*rlm - 5.7273e-1 * pow(rlm,3)

for rs = 1.0

JL = 9.9281e-1 - 2.1404*rlm + 2.1248*rlm*rlm - 8.7955e-1 * pow(rlm,3)

for intermediate values of rs linear interpolation of upper and lower values is used.

factor Rl

for rlm >= 0.0 and rlm <= 0.8

for rs = 0,

rl = 1.0235 - 4.3203 * rlm + 2.0627e1 * rlm * rlm - 6.3964e1 *

pow(rlm,3) + 1.1182e2 * pow(rlm,4) - 1.0061e2 * pow(rlm,5) + 3.6124e1 * pow(rlm,6)

for r_s = 0.25 ,

R_l = 8.5113e-1 - 2.2456 * rlm + 4.3365 * rlm * rlm - 4.8542 * pow(rlm,3) + 2.1875 * pow(rlm,4)

for r_s = 0.50 ,

R_l = 7.5814e-1-1.9554 * rlm + 2.5939 * rlm * rlm - 1.3258 * pow(rlm,3)

for r_s <= 0.75 ,

R_l = 1.0081-9.1831 * rlm + 6.2016e1 * rlm * rlm-2.4915e2 * pow(rlm,3) + 5.8633e2 * pow(rlm,4) - 7.9617e2 * pow(rlm,5) + 5.7702e2 * pow(rlm,6) - 1.7257e2 * pow(rlm,7)

for r_s = 1.0 ,

R_l = 6.016e-1 - 1.9411 * rlm + 2.8239 * rlm * rlm - 1.4813 * pow(rlm,3)

polynomials for r_s = 0.25 and r_s = 0.5 are not accurate NEAR rlm = 0, but are accurate for rlm = 0.1 to 0.8 so for rlm = 0.0, R_l = 1.0 is used separately. for intermediate values of rs linear interpolation of upper and lower values is used.

factor J_b

For Shell side Raynolds number > 100 and r_b >= 0.0 and r_b <= 0.7

for N_{ss+} >= 0.5 J_b = 1.0

for N_{ss+} = 1.0/3.0 J_b = 1.0021-1.4286e-1 * rb

for N_{ss+} = 0.2 J_b = 1.0014-2.9643e-1 * rb

for N_{ss+} = 0.1 J_b = 1.0107 - 5.3988e-1 * rb + 1.369e-1 * rb * rb

for N_{ss+} = 0.05 J_b = 1.005 - 6.6429e-1 * rb + 1.7857e-1 * rb * rb

for N_{ss+} = 0.0 J_b = 1.0036-1.197 * rb + 5.2976e-1 * rb * rb

for intermediate values of rs linear interpolation of upper and lower values is used.

$r_b = 0.0 \quad J_b = 1.0$

Recommended lower limit of J_b is given by following polynomial

J_{blimit} = 6.35e-1 + 1.0734e-1 * rb + 4.881e-1 * rb * rb-1.3889e-1 * pow(rb,3)

factor rbf

Applicability limit $r_b >= 0.0$ and $r_b <= 0.7$

for $N_{ss+} >= 0.5$ $R_{bf} = 1.0$

for N_{ss+} = 1.0/3.0 R_{bf} = 1.0086 - 4.9464e-1 * rb + 1.25e-1 * rb * rb

for N_{ss+} = 0.2 R_{bf} = 9.9786e-1-9.2202e-1 * rb + 3.1548e-1 * rb * rb

for N_{ss+} = 0.1 R_{bf} = 1.000 - 1.4018 * rb + 6.6071e-1 * rb * rb

for N_{ss+} = 0.05 R_{bf} = 9.98e-1 - 1.8014 * rb + 1.0714 * rb * rb

for N_{ss+} = 0.0 R_{bf} = 9.85e-1 - 3.175 * rb + 3.25 * rb * rb

for r_b= 0 r_{bf} = 1

factor J_c

jc = 5.3539e-1 + 6.6881e-1 * fc + 4.5468e-1 * fc * fc - 4.8668e-1 * pow(fc,3)

APPENDIX C

** SPECIFICATIONS FROM KERN METHOD **

:ONSTRUCTION DETAILS:-

Size	539.75 (mm) x 4.0 m
Number of shell passes	1
Number of tube passes	2
Number of Tubes	342
Tube Outer Diameter	19.05 (mm)
Tube Inside Diameter	16.5608 (mm) (18 BWG)
Layout Angle	30 deg.
Pitch	23.8125 (mm),
Tube Length	4 m
Baffle pitch	250 mm
Number of Baffles	15
Thermal conductivity of tube metal	112 W/m.K

THERMO HYDRAULIC DETAILS :-

		SHELL SIDE	TUBE SIDE
uid Circulated	(Kg/sec)	25	50
ecific Gravity		1	1
scosity	(N.sec/sq_m)	8.1E-4	9.1E-4
ecific heat	(J/Kg.K)	4186	4186
ermal Conductivity	(W/m.K)	0.62	0.62
mperature in	(C)	55	35

'emperature out	(C)	45	40
llowable press.drop	(N/sq_m)	1.0E5	1.0E5
alculated Press.drop	(N/sq_m)	8.225E4	2.123E4
'ilm coefficient	(W/sq_m.K)	6215	5403

lean overall heat transfer coefficient (W/sq_m.K) = 2664.7

esign overall heat transfer coefficient (W/sq_m.K) = 1100

alculated (available) Dirt Factor (sq_m.K/W) = 5.335E-4

'ube side velocity (m/sec.) = 1.3558

(BELL - TABOREK METHOD)

** Additional Specifications **

U TUBE TYPE EXCHANGER

BAFFLES: -

Single Segmantal with cut	(% of Shell ID)	= 25
Baffle Shell Diametral Clearence	(mm)	= 3.81
Baffle pitch at Inlet End	(mm)	= 375
Baffle pitch at Outlet End	(mm)	= 375
BaffleHole-TubeOD Diametral Clearence	(mm)	= 7.9375E-1
Net window flow area	(sq_m)	= 2.7337E-2
Minimum cross flow Area	(sq_m)	= 2.8575E-2
Number of Pairs of Sealing Strips	(#)	= 2
Cross flow Raynolds number	(#)	= 2.0575E+4

'RESSURE DROP

Correction factors, Rl		= 4.4826E-1
Rb		= 1.9444E-1
Rs		= 2.4018E-1
Cross Flow Pressure Drop	(N/sq_m)	= 1.1635E+4
Window Pressure Drop	(N/sq_m)	= 1.2810E+4

End zones Pressure Drop	(N/sq_m)	= 2.4180E+3
Total Shell side Pressure drop	(N/sq_m)	= 2.6864E+4

HEAT TRANSFER COEFFICINT

Ideal (Cross flow) Heat Trans. Coeff.	(W/sq_m.K)	= 8.4030E+3
Correction factors, jc		= 1.0240
jl		= 6.696E-1
jb		= 9.282E-1
js		= 9.618E-1
Cummulative correction factor		= 6.123E-1
Shell Side Heat Transfer Coefficient	(W/sq_m.K)	= 5.145E+3

** VIBRATION CHECK REPORT **

For Tube metal density = 7842 kg/ m^3

Modulus of elasticity = 2.0E11 N/ m^2

Fatigue stress = 2.0E8 N/ m^2

For Span Length = Central Baffle Pitch

Fundamental tube natural frequency	=	4.6855295889E+02 Hz
Vortex shedding Frequency	=	8.2517888941E+00 Hz
Critical cross flow velocity	=	2.7394985541E+01 (m/s)
Maximum cross flow velocity	=	8.7332494426E-01 (m/s)
Baffle damage number	=	4.3555338994E-03
Collision damage number	=	8.4101191510E-04
Maximum damage number	=	5.8792729143E-01

Safe *** in Vortex shedding

fvs/fn = 1.7611219261E-02 < 0.5

Safe *** in fluid elastic whirling

vcrit = 2.7394985541E+01 > vmax = 8.7332494426E-01

Safe *** No Baffle Damage

Nbd = 4.3555338994E-03 < Max allow number = 5.8792729143E-01

Safe *** NO Collision Damage

Ncd = 8.4101191510E-04 < Max allow number = 5.8792729143E-01

For Span Length = 2 * (Baffle Pitch)

Fundamental tube natural frequency = 1.1713823972E+02 Hz
Vortex shedding Frequency = 8.2517888941E+00 Hz
Critical cross flow velocity = 6.8487463852E+00 (m/s)
Maximum cross flow velocity = 8.7332494426E-01 (m/s)
Baffle damage number = 1.7422135597E-02
Collision damage number = 1.3456190642E-02
Maximum damage number = 5.8792729143E-01

Safe *** in Vortex shedding
fvs/fn = 7.0444877042E-02 < 0.5
Safe *** in fluid elastic whirling
vcrit = 6.8487463852E+00 > vmax = 8.7332494426E-01
Safe *** No Baffle Damage
Nbd = 1.7422135597E-02 < Max allow number = 5.8792729143E-01
Safe *** NO Collision Damage
Ncd = 1.3456190642E-02 < Max allow number = 5.8792729143E-01

For Span Length = Max. unsupported length
(First window from ends)

Fundamental tube natural frequency = 7.4968473422E+01 Hz
Vortex shedding Frequency = 8.2517888941E+00 Hz
Critical cross flow velocity = 4.3831976866E+00 (m/s)
Maximum cross flow velocity = 8.7332494426E-01 (m/s)
Baffle damage number = 2.7222086871E-02
Collision damage number = 3.2852027933E-02
Maximum damage number = 5.8792729143E-01

Safe *** in Vortex shedding
fvs/fn = 1.1007012038E-01 < 0.5
Safe *** in fluid elastic whirling
vcrit = 4.3831976866E+00 > vmax = 8.7332494426E-01
Safe *** No Baffle Damage
Nbd = 2.7222086871E-02 < Max allow number = 5.8792729143E-01
Safe *** NO Collision Damage
Ncd = 3.2852027933E-02 < Max allow number = 5.8792729143E-01